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A REVIEW OF BOILING HEAT TRANSFER PROCESSES AT HIGH HEAT FLUX

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SUMMARY

This report summarizes an extensive background review of heat transfer processes, including boiling, that are encountered in the cooling of high enthalpy arc heater components. The objectives of this report are, (1) to review the different processes encountered in backside wall cooling and identify those processes that are desirable for effective cooling of high enthalpy arc heater components, (2) to identify those parameters which affect the cooling processes of interest and the trend of cooling performance when the parameters are individually varied, and (3) to present a summary of applicable theoretical, analytical, and experimental work in surface cooling performed in the past 50 years. A significant number of parameters are found to affect the cooling processes of interest, and, although a large number of correlations exists for boiling heat transfer, considerable inaccuracy is encountered in the prediction of pertinent parameters, particularly at elevated heat loads.

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ILLUSTRATIONS

NOMENCLATURE

A	surface area, ft^2
a	speed of sound, fps
Bo	boiling number, = $q''/G\, i_{fg}$
c, c_p	specific heat, Btu/lbm °F
C_D	drag coefficient for a sphere
CHF	critical heat flux, Btu/ft^2 hr
D	diameter, ft
d	bubble departure diameter, in.
DNB	departure from nucleate boiling
erfc	complimentary error function
f	friction factor as given in Eqn. 5
G	mass velocity, = ρV except as defined in Eqn. (12), lbm/ft^2 hr
g	acceleration of gravity, ft/hr^2
g_c	gravitational constant, ft/hr^2
h	heat transfer coefficient, Btu/ft^2 hr °F
i	enthalpy, Btu/lbm
i_{fg}	latent heat of vaporization, Btu/lbm
J	energy conversion factor, = 778 ft-lbf/Btu
k	thermal conductivity, Btu/ft hr °F
L	length, ft
m_v	vapor rate, lbm/hr
Nu	Nusselt number, = hD/k
p	pressure, psia
Pr	Prandtl number, = $\mu c/k$
q'',q/A	heat flux, Btu/ft^2 hr, except as noted
q''/V	internal heat generation rate per unit volume
R	ideal gas constant
r	characteristic radius, ft
R'	Laplace number, = $r[g(\rho_l - \rho_v)/\sigma]^{1/2}$
Re	Reynolds number, = $\rho VD/\mu$
rms	root mean square
T	temperature, °F
t	wall thickness (Fig. 35)
$t_{\dot{m}/2}$	time to reach half of initial flow, sec
t_{tr}	flow rate transit time, sec
V	velocity, fps
x	vapor quality

SYMBOLS

α	thermal diffusivity, ft^2/hr
ΔC	excess concentration of the volatile component in the vapor, percent (see Fig. 17)
Δi_i	enthalpy difference due to inlet subcooling, $= i_{local} - i_{in}$, Btu/lbm
ΔT_{bulk}	complete temperature difference, $= T_w - T_b$, °F
ΔT_{sat}	superheat, $= T_w - T_{sat}$, °F
ΔT_{sub}	subcooling, $= T_{sat} - T_b$, °F
μ	viscosity, lbm/ft hr
ρ	density, lbm/ft^3
σ	surface tension, lbf/ft
τ	shear stress, lbf/ft^2 (except as defined in Eqns. 38 and 39)
Φm	specific mass flow rate in Eqn. (39), kg/m^2 sec

SUBSCRIPTS

avg	average
B	fully developed nucleate boiling
Bi	extension of the fully developed nucleate boiling curve to incipience point
b	bulk fluid
CHF	critical heat flux conditions
conv	pure convection conditions
cr	critical heat flux conditions
D	diameter
D_e	equivalent or hydraulic diameter
e	equivalent or hydraulic
f	film
FC	forced convection
h	hydraulic
htr	heater
I	inner
i,in	inlet
l	liquid
LB	subcooled boiling incipience
m	mean
O	outer
r	reduced
sat	saturation
ss	steady-state conditions
tr	transient conditions
ts	test section
v	vapor
w	wall

1.0 INTRODUCTION

The work reported herein was performed by Arnold Engineering Development Center (AEDC), Air Force Systems Command (AFSC), under Program Element 65807F. The Air Force Program Managers were Capt. D. G. Burgess, Capt. H. Martin, and Lt. P. Zeman, DOT. The work was performed by Calspan Corporation/AEDC Operations, operating contractor for the Aerospace Flight Dynamics testing effort at AEDC, AFSC, Arnold Air Force Base, Tennessee. The work was performed in the Aerospace Systems Facility (ASF) under AEDC Project Number DD01. The work was conducted during the period between 1 October 1990 and 30 November 1991.

Water cooling is used extensively at AEDC to prevent the failure of high enthalpy arc heater components at elevated heat loads where cooling by radiation, conduction, and natural convection from air circulation are ineffective. Other cooling techniques such as the use of liquid metals, transpiration or film cooling pose their own set of disadvantages or limitations (Ref. 1) and are certainly possibilities for providing cooling at much more severe heating conditions where water cooling alone may prove to be inadequate. The current interest is in exploiting backside water cooling to the limit of its capability, especially for arc heater nozzles. Such an application requires an understanding of the parameters that affect the cooling processes and the determination of the limiting point[1] where the surface fails due to inadequate cooling.

The purpose of this report is threefold, (1) to review the different processes encountered in backside wall cooling and identify those processes that are desirable for effective cooling of high enthalpy facility components, (2) to identify those parameters which affect the cooling processes of interest and the trend of cooling performance when the parameters are individually varied, and (3) to present a summary of applicable theoretical, analytical, and experimental work in surface cooling performed in the past 50 years.

2.0 THE BASIC BOILING CURVE

The characteristic temperature response of a surface cooled by an adjacent fluid as heat flux is increased was first proposed by Nukiyama (Ref. 2) in 1934. Figure 1 shows the typical shape of the curve, known as the boiling curve, obtained when heat flux from a cooled surface is plotted as a function of the liquid superheat (the difference between the surface temperature and the liquid saturation temperature). At low heat flux (segment A-B in Fig. 1) the surface temperature is below the saturation temperature of water and similar liquids at or above atmospheric pressure. Cooling at low heat flux is by pure convection heat transfer and is fairly well understood. In this region, analytical solutions and experimental correlations

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[1] The limiting point is also identified as the burnout point, boiling crisis, departure from nucleate boiling (DNB), or critical heat flux condition among other terms by various authors. Terminology for the limiting point will be discussed under the section entitled The Basic Boiling Curve.

(for more complex flow geometries) have been derived which allow for accurate prediction of heat transfer in various cooling configurations (see, for example, Chapters 8-9 of Lienhard Ref. 3, Chapter 6 of White Ref. 4, or Kays and Crawford Ref. 5, Chapters 13-14). As heat flux from the surface is increased beyond the point where the surface temperature becomes significantly greater[2] than the water saturation temperature (point B in Fig. 1), vapor generation, better known as boiling, occurs from discrete nucleation sites on the surface or in the bulk fluid depending on the bulk fluid temperature. The region of boiling from discrete sites is termed nucleate boiling and is shown as segment B-C in Fig. 1.

As the upper limit of nucleate boiling is reached (point C in Fig. 1), an undesirable transition to another form of boiling, known as film boiling, begins. Because the boiling near the limit of nucleate boiling becomes so violent, numerous vapor bubbles begin to combine near the surface. Since the vapor has a much lower thermal conductivity than the liquid, the patches of vapor effectively insulate the surface from the efficient heat transfer to the liquid as experienced during nucleate boiling. Unlike a temperature driven system which would proceed towards point D[3] in Fig. 1, a heat flux driven system such as a high enthalpy arc facility would proceed directly towards point E[4]. The processes involved when a boiling system undergoes the path from the limit of nucleate boiling (point C) towards stable film boiling (point E) are highly unstable, and failure of the surface due to melting occurs in most cases because the melting temperature of the surface material is lower than the temperature associated with stable film boiling at point E. The point at which transition to film boiling begins is often termed the burnout point even though in some cases failure does not occur (e.g., a temperature driven system or a surface material with a very high melt temperature). The burnout point is known also as the departure from nucleate boiling (DNB), fast burnout and burnout of the first

2 A certain degree of liquid superheat is required to establish a flow of heat to create vapor bubbles. The superheat can become quite large for very smooth surfaces or highly wetable fluids (e.g., refrigerants or liquid metals) which can lead to explosive boiling (known as bumping) that can lead to structural damage to hardware.

3 Transition boiling occurs between the limit of nucleate boiling (point C) and point D which is known as the Leidenfrost point. The path between the points is dependent on numerous factors and the heat flux becomes a multivalued function of the liquid superheat (e.g., see Witte and Lienhard, Ref. 6). This region is inaccessible with a heat flux controlled system.

4 Point E lies on a portion of the boiling curve known as film boiling. Beginning at the Leidenfrost point (point D) heat transfer across the vapor film adjacent to the surface increases due to radiation from the wall, through the vapor, to the bulk fluid.

kind in systems which involve a subcooled[5] liquid. In saturated liquid boiling systems another type of burnout occurs (usually at lower heat flux than subcooled boiling) and the terms slow burnout, dryout, or burnout of the second kind are used (see Ref. 8). Boiling crisis and critical heat flux condition are used in a general sense to describe all burnout situations. Because of a lack of standard nomenclature, hereafter the point at which transition to film boiling begins is designated the burnout point or boiling crisis, and the peak, ultimate, or maximum heat flux is termed the critical heat flux (CHF) except where designated differently in the referenced material.

Design of the cooled components of a high enthalpy facility for safe operation necessitates the cooling processes to lie to the left of, and well below, point C in Fig. 1. High heat transfer coefficients, hence, efficient energy transfer, can be realized when operating in the nucleate boiling regime rather than the pure convection regime at low to moderate coolant pressure and velocity. Any method used to increase the CHF will provide a factor of safety when operating well into the fully developed nucleate boiling regime. The determination of the CHF along with the shape of the nucleate boiling curve for a given cooling system remains one of the most challenging areas of research despite the considerable attention it has received in the last 50 years (Ref. 9). Complexity of the boiling mechanisms, their unsteady nature, and the small scale of the heat transfer processes have hindered the development of theoretical models and understanding of the phenomenon from experimental studies. Moreover, the cooling flow structure and, hence, the heat transfer processes vary depending on the heating condition, fluid properties and surface geometry. For instance, boiling heat transfer that occurs in a high enthalpy facility (subcooled flow, surface boiling optimized for effective wall cooling over short lengths at high heat loads) can be quite different than that which occurs in a nuclear reactor (saturated flow with possible slug or annular two phase flow, bulk boiling optimized for efficient energy transfer over great lengths at low to moderate heat loads).

The objective of a large portion of the experimental work undertaken in recent years in the area of boiling heat transfer has been to obtain an empirical or semiempirical correlation for a given cooling configuration and set of test conditions. Many of the correlations have limited application and cannot be generalized for use in analyzing other cooling configurations and conditions. The experiments do provide a qualitative evaluation of what parameters affect the boiling processes of interest and the trend in cooling performance when each of the

[5] If the cooling fluid bulk temperature is well below the saturation temperature of the fluid, then the fluid is known as subcooled. Boiling that occurs on a surface adjacent to such a fluid is known as subcooled, local or surface boiling. If the bulk temperature is at or near the saturation temperature, the fluid is known as saturated. Bulk boiling can occur when the fluid is near saturation, and undergoes various flow regimes in forced flow such as slug, annular, and drop flow. See Collier (Ref. 7) for further discussion on saturated flows.

parameters are varied. Optimizing the influential parameters for high enthalpy facility component cooling will, in turn, aid in identifying those boiling mechanisms which play a part in the cooling process of interest. Once the mechanisms are identified, then appropriate analytical models and experimental data may be assembled.

3.0 PARAMETRIC EFFECTS ON THE BOILING CURVE

More than 20 parameters have been identified that affect the cooling process. Some parameters play an important role from pure convection to boiling crisis while others play intermittent roles. For instance, velocity has a large effect on cooling performance in the pure convection regime and at boiling crisis, but it has virtually no effect in the nucleate boiling regime. Therefore, the contribution of each parameter in each of the three regions of interest (pure convection, nucleate boiling, boiling crisis) will be discussed. The effects of various parameters on the microscopic boiling scale (i.e., bubble characteristics) have not been studied in great detail and are presented here for the more influential parameters. It should be pointed out that the cooling process is not without parametric distortion, that is, the change in one parameter causing a change in another parameter. Because of the complexity of the boiling process, many of the parametric distortions are ill-defined or unknown; therefore, a discussion of these distortions included here is limited. Several excellent reviews of various parametric effects on boiling are available and are drawn upon in the following review, most notably: boiling in general - Tong (Ref. 10), Westwater (Refs. 11 and 12), Rohsenow (Refs. 13 and 14), Addoms (Ref. 15), Katto (Ref. 16); convective boiling - Collier (Ref. 7); and subcooled flow boiling - Boyd (Refs. 1 and 17), Bergles (Ref. 8), Macbeth (Ref. 18), Hewitt (Ref. 19), Fiori and Bergles (Ref. 20), Hughes (Ref. 21).

3.1 IMPORTANT PARAMETERS AND THEIR EFFECTS ON HEAT TRANSFER

Velocity-

Velocity (or mass velocity, G, when combined with the fluid density), as mentioned previously, plays an intermittent role in the cooling process. Figure 2 (from Kreith and Summerfield, Ref. 22) illustrates the significant effect of fluid (butanol) velocity as heat flux from an adjacent surface increases in pure convection heat transfer. However, as transition to boiling occurs the effect of velocity diminishes. Clark and Rohsenow (Ref. 23) and Tong (Ref. 10) among others have noted that velocity has little or no effect on heat transfer in vigorous (or fully developed) nucleate boiling. Boiling which occurs on a surface or in a fluid initially at rest and only free convection motion is present is known as pool boiling. Boiling which occurs on a surface or in a fluid where the fluid has a bulk (imposed by some external source such as a pump) and free convection motion is known as forced convection boiling or simply flow boiling. The advantages of flow boiling over pool boiling, when present in a cooling process, become evident as boiling crisis is approached. McAdams, et al. (Ref. 24) showed that the CHF could be increased with increasing velocity (Fig. 3). Grace (Ref. 25) generalized the velocity effect on CHF for various qualities of water liquid/vapor systems as shown in Fig. 4. An inversion in velocity effectiveness occurs at higher vapor qualities (i.e., saturated fluid) as illustrated in Fig. 4; however, as will be discussed next, subcooling (vapor qualities less than zero) provides the largest values of CHF, where increasing mass velocity increases the CHF for a given degree of subcooling (Fig. 4).

Analyzing the boiling process on a microscopic scale, Abdelmessih, et al. (Ref. 26) found that both the maximum bubble radius and the bubble life span decrease as the liquid velocity increases (Fig 5). Gunther (Ref. 27) found in addition to decreasing the bubble radius and lifetime, that increasing velocity also decreases the bubble population. Tippets (Ref. 28) found that at high velocity combined with large subcoolings that the flow adjacent to a surface sustaining boiling is an irregular frothy layer made up of bubbles that grow, slide, and then collapse along the surface. These bubble characteristics make their detection and measurement difficult at high flow velocity.

Subcooling-

McAdams, et al. (Ref. 24) showed that subcooling has a slight effect on heat transfer in the pure convection and the nucleate boiling regions of the boiling curve due to small variations in physical properties of the fluid at different temperatures. Subcooling, like velocity, has its primary advantages at boiling crisis. Van Huff and Rousar (Ref. 29) and Kreith (Ref. 30) indicated that subcooling and velocity are the most important variables on burnout (CHF). Green, et al. (Ref. 31) measured burnout heat flux for vertical water flow in a rectangular channel at high pressure and found that increasing both velocity and subcooling increased the burnout heat flux (Fig. 6). Van Huff and Rousar (Ref. 29) combined the two variables and derived a linear fit to previous data (Fig. 7). Macbeth (Ref. 18) found that increasing subcooling increased CHF linearly at constant mass velocity; however, Katto and Yokoya (Ref. 32) indicated that nonlinear regimes can occur at high mass velocity at low to moderate subcooling, and Lienhard (Ref 33) found that the maximum heat flux (CHF) versus degree of subcooling curve "flattens" out for any velocity above a threshold degree of subcooling. In any event, it is likely that the CHF generally can be increased by increasing velocity and subcooling.

On a microscopic scale, Gunther (Ref. 27), using a flow boiling apparatus, found that the bubble lifetime, surface coverage, and maximum radius decreased with increased subcooling, while the bubble population, which initially decreases, increases for subcoolings above 130 °F (Fig. 8). Dew (Ref. 34) confirmed Gunther's findings except bubble population, which decreased with increased subcooling. Ibrahim and Judd (Ref. 35) confirmed that the growth period decreased with increased subcooling and, in addition, found that the waiting period between bubble generations from an active nucleation site also decreased with increased subcooling except at low subcooling (below 40 °F). It should be noted, however, that Ibrahim and Judd performed their subcooling studies on a pool boiling apparatus.

Pressure-

The primary contribution of pressure on a cooling process is its effect on the saturation temperature of the fluid, although an interesting inversion of pressure effect occurs at boiling crisis. Kreith and Summerfield (Ref. 22), who studied flow boiling with butanol, found that increasing pressure had no effect on heat transfer in the pure convection region but increased wall temperature for a given heat flux in the nucleate boiling region as shown in Fig. 2. An increase in wall temperature is not particularly beneficial; however, some improvement to the CHF can be achieved by increasing pressure - up to a point. Cichelli and Bonilla (Ref. 36), while studying pool boiling of organic liquids, found that heat flux increased with increasing pressure to a maximum value which occurred at about one-third of the critical pressure (Fig. 9). Addoms (Ref. 15) obtained similar results from pool boiling of water; however, the maximum peak heat flux varied with heater wire size (Fig. 10). Boyd (Ref. 1) suggested a similar maximum CHF (Boyd termed it the optimum CHF) in flow boiling that occurs at various pressures depending on mass velocity and subcooling (Fig. 11).

Tolubinsky and Kostanchuk (Ref. 37) found that increased pressure decreased the maximum bubble diameter and the generation frequency (Fig. 12), although the former appears to remain constant or to begin increasing above a pressure of 8 to 10 bar (120 to 150 psia). The bubble life span is inversely related to the generation frequency, and, therefore, increases with increased pressure.

Surface Roughness-

Surface roughness plays a complicated role in cooling efficiency and is divided into micro-roughness (typically smaller than 120 μin.) and macro-roughness (artificial protrusions such as knurling, fins, or ribs). Corty and Foust (Ref. 38) performed an extensive micro-roughness study of various fluids in pool boiling on copper and nickel surfaces. They found that increased roughness had no effect in the pure convection region but heat transfer coefficients were more than doubled in the nucleate boiling region for the roughest surface when compared to the smoothest surface (Fig. 13). Brown (Ref. 39), however, found a modest effect of surface finish in subcooled, low velocity flow boiling at moderate heat flux (Fig. 14). Berenson (Ref. 40) and later Ramilison and Lienhard (Ref. 41) found a modest surface finish dependence of boiling burnout (up to approximately 20 percent decrease from smooth to rough surface) in pool boiling. Other than Weatherhead (Ref. 42) who concluded that CHF is decreased for very smooth surfaces, other investigators such as Leung, et al. (Ref. 43), Aladyev, et al. (Ref. 44), and Bergles and Morton (Ref. 45) found little or no effect of surface microroughness on CHF in flow boiling. Bergles and Morton further determined that any improvements of the burnout heat flux caused by roughness appear to diminish with increasing velocity and subcooling.

Macro-roughness has been found to have a more pronounced effect on forced flow cooling efficiency than the micro-roughness previously discussed. Types of macro-roughness used in forced flow include knurling (Durant, et al. - Ref. 46), fins (Kovalev, et al. - Ref. 47, and Shim, et al. - Ref. 48), and ribs (Ravigururajan and Bergles - Ref. 49, and Akhanda and James - Ref. 50). Durant and his team determined that liquid-film heat transfer coefficients of knurled surfaces were up to 75 percent higher than those of smooth surfaces. Burnout heat flux was shown to increase by as much as 80 percent (Fig. 15). The types of roughness investigated by Durant's team were comparatively large (greater than 0.005 in.) over those discussed in the previous paragraph. Macro-roughness is considered a heat transfer enhancement technique rather than a basic parameter affecting heat transfer that would be inherent in a given system. An excellent review of this enhancement technique is presented by Webb (Ref. 51); however, most of the surface geometries have been developed for pool boiling only.

Coolant Properties-

The understanding of the effects of coolant properties on the heat transfer processes in boiling remains limited even though selection of the fluid is a most important consideration in boiling applications (Ref. 1). The physical properties of a given fluid such as liquid/vapor density ratio, latent heat of vaporization, surface tension, and heat capacity, and transport properties (i.e., viscosity and thermal conductivity) may play a role in the cooling process; however, distinguishing their individual contribution is difficult and little or no data exist to ascertain their effect. The properties are related to the pressure and/or the temperature of the fluid, and the problem of parametric distortion discussed earlier becomes evident.

The limited studies of the microscale mechanisms during boiling have produced conflicting information as to the effect of latent heat. A number of investigations (Refs. 52-55) has revealed that latent heat transport accounts for as much as 50 percent of the measured heat flux in subcooled boiling; however, Forster and Grief (Ref. 56), Brown (Ref. 39), and Plesset and Prosperetti (Ref. 57) found very little latent heat contribution.

Pike, et al. (Ref. 58) found using a pool boiling apparatus that increasing viscosity hindered the onset of boiling (decreased heat transfer coefficient, increased wall superheat) and lowered the CHF. As will be discussed under transient operation effects, it has been proposed that transient operation is viscosity dependent, thereby contributing to an observed difference in CHF for the transient and steady-state heating of certain fluids.

Surface tension may or may not play a beneficial role in the cooling process (Ref. 11). Fiori and Bergles (Ref. 20) identified several investigations that clearly showed that CHF decreases with decreasing surface tension. The addition of various amounts of surface-active agent has been found to increase the heat transfer quite significantly at the expense of reducing the CHF. For example, Frost and Kippenhan

(Ref. 53) showed that for a given wall temperature the heat flux increased by approximately 50 percent when a surface-active agent was added to water thereby reducing the surface tension (Fig. 16). The reduction of surface tension had a detrimental (decreasing) effect on the CHF, which agrees with previous results, and is evident in Fig. 16.

Improvements to the CHF have been demonstrated in a limited number of investigations using binary mixture components. Papaioannou and Koumoutsos (Ref. 59) and Wei and Maa (Ref. 60) found that polymeric additives to water increased heat transfer coefficients and broadened the peak (CHF) of the boiling curve. Tolubinskiy and Matorin (Ref. 61) found that for ethanol-water and acetone-water mixtures the maximum CHF (occurring at 20 percent ethanol/80 percent water and 10 percent acetone/90 percent water) exceeded the CHF for water by 30 and 40 percent, respectively (Fig. 17).

A number of coolants such as refrigerants, cryogenics, organics, binary mixtures, liquid metals, and propellants in addition to water have been investigated for various cooling applications. A propellant is probably not attractive for use as a coolant in a high enthalpy facility except in specific test article configurations. The use of organics is unlikely, except possibly as a trace additive or in a binary mixture, because of their lower performance compared to that of water (Ref. 62) and their tendency to decompose at surface temperatures experienced at nucleate boiling heat flux levels (Ref. 8). Binary mixtures, as discussed previously, may be an attractive approach to improve the CHF in a water-cooled system. Liquid metals may be considered in the future as an alternative or compliment to water cooling. Because of their high thermal conductivity, liquid metals have proved useful in convective cooling applications. Boiling in liquid metals, however, is somewhat confused. Lyon, et al. (Ref. 63) showed that mercury, for instance, appears to enter film boiling at surprisingly low wall superheats. And because of the high wettability of liquid metals, high superheats are normally required to initiate boiling, in some cases having an explosive transition that can cause structural damage to hardware. A thorough understanding of boiling metals is needed prior to their application in high heat flux/cooling situations. Refrigerants such as freons have been used in previous experiments (Refs. 64-67) to simulate water cooling at high heat flux conditions since the CHF can be achieved at much lower heat flux. Scaling methods are applied to the simulation to achieve high heat flux/water cooling predictions (see, for example, Ahmad, Ref. 68). A good example of this "fluid-to-fluid" scaling is presented by Bergles in Ref. 8. Use of refrigerants for actual cooling in high enthalpy facilities is not recommended because of their low thermal conductivity and CHF levels. In addition, refrigerants suffer from a high wettability problem not unlike that of liquid metals. Similarly, Powell (Ref. 69) found that low heat transfer coefficients and CHF levels existed for cryogenics such as hydrogen, nitrogen, and oxygen which would limit the use of liquified gases as a heat transfer fluid. Refrigerants and cryogenics perform well in systems that have relatively low heat flux requirements and limited space availability.

Wall Material-

Though considered a secondary effect, there appears to be a contribution of surface material properties on cooling efficiency. Early experiments in pool boiling demonstrated a difference between stainless steel and copper in nonboiling and boiling water (Ref. 70); copper, gold, and chromium in boiling ethanol (Ref. 71); nickel, tungsten, and chromel in boiling water (Ref. 72); and aluminum, copper, iron, and chrome plated copper in water and various organic liquids (Ref. 73) among others. Later, Magrini and Nannei (Ref. 74) found that the heat transfer coefficient appears to be dependent on the thermal activity, $\sqrt{\rho c k}$, for heaters of different metals above a limiting value of thickness (Fig. 18). Kovalev, et al. (Ref. 75) showed that copper with coatings of low thermal conductivity tended to increase the degree of wall superheat over that of clean copper boiling in Freon 113. Cheng and Ragheb (Ref. 76) found that a greater wall superheat was required for an Inconel surface than a copper surface in flow boiling of water (Fig. 19).

Numerous pool boiling experiments have demonstrated that the CHF has a significant material dependence (Refs. 71-73 and 75) which has not been observed in flow boiling. The data of Cheng and Ragheb presented in Fig. 19 show the CHF to be equivalent for Inconel and copper. Jacket, et al. (Ref. 77) found no effect on CHF due to changing wall material from nickel to Zircaloy or, later, stainless steel (Ref. 31) in flow boiling of water at high pressure. Van Huff and Rousar (Ref. 29) identified studies where various metals had no effect on CHF except where the material and coolant were incompatible (additional reactions occur). Never-the-less, Fiori and Bergles (Ref. 20) suggest that materials with high thermal diffusivity (e.g., aluminum and molybdenum) are expected to have higher CHF which was later confirmed by Knoebel, et al. (Ref. 78).

Wall Thickness-

The thickness of the heated wall has been shown to have an effect on the boiling heat flux, but only below a critical or limiting value of wall thickness. Magrini and Nannei (Ref. 74), for instance, found for saturated pool boiling of water that the limiting value was approximately 15 μm (0.0006 in.) for tin and nickel, and negligible for copper and silver. They also determined the limiting value for zinc to be approximately 70 μm (0.0028 in.), although it is not obvious from the data presented in Ref. 74. Del Valle and Kenning (Ref. 79), while studying subcooled water flow at high heat flux, found that thin heated walls (0.003 to 0.008 in. thickness) had no effect in the nonboiling region, but the rate of nucleate boiling heat transfer at a given wall superheat increased with increasing wall thickness (Fig. 20). Tippets (Ref. 28) found that CHF measured for 0.006-in. thick heater ribbons were approximately 20 percent lower than for 0.010-in. thick ribbons in saturated water flow. They suggest that thicker walls can maintain a higher internal heat generation rate per unit volume (q''/V) before the critical condition is reached than thin walls since the latter will have a larger temperature rise at a given heat flux (i.e., q''/V will be greater for a thin walled tube since the volume is smaller and the heat flux constant). Fiori and Bergles (Ref. 20), using subcooled water in a flow boiling apparatus, showed that thicker walled tubes (0.078 in. versus 0.012 in.) had a CHF up to 58 percent higher than thin walled tubes. Aladyev, et al. (Ref. 44), however, found no effect of wall thickness (0.016 to 0.079 in.) on the CHF for water flow boiling.

<u>Geometry-</u>

Geometric considerations that have been experimentally shown to have an effect on the cooling process include the diameter (or hydraulic diameter) of the heater (or channel), heated length (or L/D), the channel cross section and longitudinal configuration, and the flow orientation (Ref. 1). Because of various applications, experiments performed in the past associated with studying cooling problems have encompassed a wide range of configurations, and one must be cautious when comparing data from one experiment to another (Ref. 11).

The channel diameter appears to have a varied effect on boiling heat transfer. Schweppe and Foust (Ref. 80) found that for flow boiling with low velocity water that the wall superheat increased (or heat transfer coefficient decreased) as channel diameter increased from 0.438 to 1.049 in. Collier (Ref. 7) presented subcooled, flow boiling data of Lee and Obertelli (Ref. 81) and Matzner (Ref. 82) that indicated that CHF increased as channel diameter increased from 0.22 to 1.475 in. at constant subcooling (Fig. 21). Bergles (Ref. 83), however, found for subcooled flow boiling that CHF increased as channel diameter decreased from 0.33 to 0.023 in. (see Fig. 22), although the percentage increase in CHF is somewhat less when parametric distortion is accounted for (Ref. 20). Van Huff and Rousar (Ref. 29) found no effect of channel diameter on CHF for subcooled flow boiling with diameters ranging from 0.107 to 0.587 in.

Schweppe and Foust (Ref. 80), by varying channel diameter (as previously discussed) and holding the heated length constant, showed for flow boiling with low velocity water that the wall superheat increased as L/D decreased below 15 to 20. Bergles (Ref. 83) found for flow boiling that CHF increased as L/D was decreased below 35 (Fig. 23). Similarly, Ornatskiy (Ref. 84), using subcooled water flowing in circular ducts, found this limiting L/D to be between 20 to 24, and Jens and Lottes (Ref. 85) showed that, for high pressure, subcooled flow boiling, changing L/D from 110 to 21 had an increasing effect on CHF. Lee and Obertelli (Ref. 81) found a considerable effect on CHF in subcooled, low velocity flow for L/D values less than 300 (Fig. 24). Van Huff and Rousar (Ref. 29), however, noted that data acquired by Aerojet General Corporation for high velocity, highly subcooled water, showed very little effect of L/D (ranging from 13 to 100) on CHF. As Boyd (Ref. 1) points out, since the L/D limit is related directly to the flow development, this limit is not a constant and is related to the flow parameters and fluid properties.

The geometric considerations thus far discussed have been for circular tube configurations. External flow over rods (or wires) and rod bundles, jets, internal flow in multiple tube (or tube bundle) configurations, rectangular channels, and, to a lesser extent, annuli have been studied in the past. Internal tube flow and annular flow closely simulate the cooling configurations encountered in a high enthalpy arc facility. Becker and Hernborg (Ref. 86) found for water flowing in an annulus with inlet subcooling and an inner heated wall (similar to an arc facility nozzle) that CHF data were considerably below those for comparable flow in a circular tube, and Zenkevich, et al. (Ref. 87) also found a difference in CHF between annuli and tubes (at the same length and diameter). Barnett (Ref. 88), however, points out significant difficulties which arise when comparing tube and annulus data and the definition of an appropriate equivalent diameter. In addition, Alekseev, et al. (Ref. 89) found significant heated length effects for annuli at much greater lengths than for circular tubes. Collier (Ref. 7) discusses other characteristics of annular flow of interest. Specifically, eccentricity of the internal heated wall will tend to reduce the CHF (see also Ref. 88). In addition, Collier points out that the outer wall in a concentric annulus system has little or no effect on the CHF for the inner surface.

The channel longitudinal configuration is important to the boiling heat transfer when significant curves or twists are introduced. Such longitudinal changes have an effect on the acceleration and turbulence of the coolant flow. A discussion of these types of configurations is included in the section entitled Acceleration and Turbulence.

Coolant flow orientation (horizontal, inclined, or vertical) has been shown to have an effect on cooling efficiency due primarily to buoyancy (gravity) causing stratification of the liquid/vapor flow (Refs. 90 and 91). The effect of flow orientation, though, diminishes with increasing subcooling and/or mass velocity, and there appears to be a threshold above which no effect of orientation has been observed (Refs. 1, 7, 91, and 92). Zeigarnik, et al. (Ref. 93), for instance, showed that the orientation of the heater plate in a rectangular flow channel had little effect on the CHF at mass velocities above 9000 kg/m^2 sec (1850 lbm/ft^2 sec) in subcooled water flow (Fig. 25). Boyd, et al. (Ref. 94) suggests that the mass velocity limit is between 3000 and 4500 kg/m^2 sec (600 to 900 lbm/ft^2 sec), which agrees with the velocity limit of 4000 kg/m^2 sec (820 lbm/ft^2 sec) found by Merilo (Ref. 91).

<u>Gas Content-</u>

A number of experiments in pool boiling (Refs. 58 and 95) and flow boiling (Refs. 21, 24 and 96) have demonstrated the effect of dissolved gases on pure convection and nucleate boiling heat transfer. McAdams, et al. (Ref. 24), for example, found little effect of dissolved air on forced convection heating, but the presence of air tended to reduce the wall superheat necessary to initiate and sustain nucleate boiling (Fig. 26) which is typical[6] of data obtained in boiling experiments where the effect of dissolved gas was evaluated. In several experiments, the effect

[6] Only Jens and Lottes (Ref. 85) found that boiling incipience is *not* affected by gas content.

of dissolved gases was found to diminish in fully developed nucleate boiling (Refs. 24 and 97). This is evident in Fig. 26 at high wall superheat where the data for boiling with dissolved air match that for degassed boiling.

Data obtained to evaluate the effect of dissolved gases on CHF are contradictory; however, in experiments where an effect was noted (Refs. 24, 96-98), the presence of dissolved gases tended to reduce the CHF from that of degassed boiling at the CHF. Fisenko, et al. (Ref. 96) found this reduction to be as high as 23 percent (at higher subcoolings). A number of investigators (Refs. 21, 31, 99 and 100) identified no appreciable effect of dissolved gases on CHF. Kalayda, et al. (Ref. 101) found that for water flow boiling the gas content had little effect on CHF at pressures below 10 MPa, but at higher pressures the CHF was reduced with increased concentrations of nitrogen.

Other experimental results have been obtained with gassed and degassed coolants. Jens and Lottes (Ref. 85) and Buchberg, et al. (Ref. 97) in water flow boiling found that the presence of dissolved nitrogen had a small effect on the pressure loss along the heated wall. Baranenko, et al. (Ref. 102) found for atmospheric pool boiling of water that the bubble diameters are larger when degassed water was used rather than gassed water. It should be noted that the presence of dissolved oxygen can have an additional effect on the heat transfer process due to oxidation of surfaces in contact with the coolant. A discussion of this oxidation effect is included in the following section.

<u>Surface Aging, Deposits, and Coatings-</u>

Surface aging[7] has been shown in the past to increase the wall superheat in pool boiling (Refs. 11 and 13). Bonilla and Perry (Ref. 71), for instance, found this to be the case on a chrome-plated copper heater with boiling ethanol. Hughes (Ref. 21) found for subcooled flow boiling that surface aging tended to decrease the CHF, and Akhanda and James (Ref. 50) found that aging increased the wall superheat in forced convection boiling of water.

The effect of surface deposits (or fouling) on cooling heat transfer are much more difficult to ascertain. Epstein (Ref. 103) identified several categories in which fouling (primarily in forced flow heat exchangers) can occur. These include precipitation (dissolved substance deposits), particulate (suspended particle deposits), chemical reaction (surface is not a reactant) and corrosion (surface reactions) among others. Collier (Ref. 7) pointed out that porous deposits on heat transfer surfaces can influence the CHF both favorably and adversely (see also Westwater, Ref. 11). Bui and Dhir (Ref. 104), found that the presence of an oxide increased the CHF in pool boiling of water. Mel'nikov, et al. (Ref. 105), however, found that CHF decreased nearly 36 percent (at a quality of 0.06) with the presence

7 Nucleate boiling on clean surfaces will not reach a steady state condition for a sizable period of time (in some cases, an hour or more) due to several reasons including surface outgassing, chemical reactions, etc. (Ref. 11). This process of reaching steady state conditions is known as aging.

of a 55-μm (0.002-in.) thick ferric-oxide scale as compared to a scale-free surface in forced-convection boiling of water.

The effect of surface coatings (primarily used for heat transfer enhancement) on cooling heat transfer is similar to that of deposits. For example, Zhukov, et al. (Ref. 106) found that the wall superheat and CHF primarily decreased for an aluminum oxide coating but both increased for an enameled surface in Freon pool boiling. As discussed previously in the wall material effects, Kovalev, et al. (Ref. 75) found that low thermal conductive coatings increased the wall superheat. Collier (Ref. 7) discusses improvements in heat transfer demonstrated by several investigators using thin Teflon films.

Acceleration and Turbulence-

High enthalpy facility components invariably require coolant flow direction and area changes which, in turn, cause flow acceleration/deceleration and turbulence. Merte and Clark (Ref. 107) found for pool boiling of saturated water that increasing the acceleration (g-loading of 1 to 21) decreased the wall temperature and thereby increased the heat transfer coefficient in both the pure convection and nucleate boiling regions, although the effect of acceleration diminished as the boiling became fully developed. Forced convection acceleration/turbulence effects have been studied with curved channels and coiled tubes. Gu, et al. (Ref. 108), studying a fluorocarbon liquid flowing over a concave shaped heater, found that for the same flow velocity and subcooling, both the heat transfer coefficient and CHF are higher in a curved channel (concave section) than in a straight channel. Gu's team also found that the increase in CHF is significant (40 percent higher than straight channel CHF) at higher flow velocities. Hughes (Ref. 21) found similar results for concave heater surfaces in subcooled Freon 113 flow boiling. Hughes found that the CHF was increased up to 50 percent in concave channels over straight channels (Fig. 27). Hughes' data, however, illustrate the disadvantages of such a curved channel configuration. Should the channel be heated entirely, the convex portion could have a detrimental effect on CHF as shown in Fig. 27. The CHF for convex surfaces were as much as 22 percent lower than that for a straight channel. Jensen and Bergles (Ref. 92) found low CHF levels in subcooled R-113 flow over inside surfaces (convex portion) of helically coiled tubes, and Bergles (Ref. 8) noted similar results found by other investigators. Winovich and Carlson (Ref. 109) made use of the advantages of a concave surface in the design of an undulating flow channel used in various high enthalpy facility components. They found that the undulating channel significantly increased (as much as 100 percent) the CHF over that of a straight channel (Fig. 28).

Swirl flow devices and other inserts have been used in the past to induce turbulence thereby enhancing the cooling heat transfer in applications such as heat exchangers. Kudryavtsev, et al. (Ref. 110), using a twisted tape in tubes with saturated water flow boiling, showed that the turbulence, generated by density gradients and acceleration from the twisted tapes, decreased the rate of nucleate boiling (increased wall superheat). However, Lopina and Bergles (Ref. 111) found little effect of swirl on the wall superheat for subcooled water flow boiling. Ornatskiy, et al. (Ref. 112) demonstrated that induced swirl at the inlet of a concentric annulus increased the CHF by 10 to 60 percent over unswirled flow for low qualities. Gambill, et al. (Ref. 113) found similar results for subcooled water flow boiling. Ornatskiy's team also showed that the effect of swirl on CHF decreases with increasing pressure (Fig. 29), but increases with increasing mass velocity (Fig. 30).

Megerlin, et al. (Ref. 114) augmented the heat transfer in subcooled water flow in tubes by using brush and mesh inserts, showing that heat transfer coefficients up to nine times the coefficients of empty tubes can be obtained. It should be pointed out that Megerlin and his team found that the inserts produced very large pressure drops and very large wall superheats. Bergles (Ref. 8) recommends that the inserts are more suitable for use in single-phase flow heat transfer augmentation. As mentioned previously in the discussion of surface roughness effects, macro-roughness can be used to enhance the heat transfer in forced flow, which probably involves some type of turbulence mechanism.

__Heat Flux Distribution-__

Variations in heat flux distribution, both axial and circumferential, have been shown to have an effect on boiling heat transfer, primarily on the CHF. Stein and Begell (Ref. 115), using subcooled water flowing in internally heated annuli, found no significant effect of an axial heat flux distribution (cosine shape) on the heat transfer coefficients. Various researchers, though, have found significant effects when CHF is approached. Styrikovich, et al. (Ref. 116), while studying subcooled and saturated water flow in pipes with uniform and linearly varying heat flux, found that the CHF for a linearly increasing heat flux (along the tube length) was nearly twice that of a tube with uniform heat flux for low quality water at 100 atm (mass velocity of 410 lbm/ft^2 sec) and 40 to 50 percent higher at 180 atm. For these cases the crisis always occurred at the exit of the heated tube. Styrikovich's team also found that the CHF for a linearly decreasing heat flux occurred near the inlet of the heated section and was larger than that with a uniform heat flux, although data errors prevented the increase from being quantified for the subcooled case. Ornatskiy, et al. (Ref. 117) found similar results for subcooled water flow in annuli with linearly or parabolically increasing heat flux; however, they found quite different results for other nonuniform heat flux applications. Earlier (Ref. 112), Ornatskiy's team had shown that for a cosinusoidal heat flux distribution on annuli with subcooled water flow the CHF can be as much as 80 percent lower than that of a uniformly heated annulus at similar conditions, and the difference between the nonuniform heating CHF and uniform heating CHF increased with a decrease in pressure and mass flow rate. In addition, data presented in Ref. 117 show that, for subcooled water flow in annuli, the CHF with sinusoidal, exponential, or linearly/parabolically decreasing heat flux is generally lower than the CHF with uniform heat flux. Similar results were presented by Zenkevich, et al. (Ref. 87) for subcooled and saturated water flowing in tubes with uniform and cosinusoidal heat flux distribution, though at higher pressure and subcooled conditions the difference appears to diminish or even reverse (Fig. 31). They point out that for complex nonuniform heat flux applications the location of the burnout point along the length of tube or annulus is difficult to predict. Swenson, et al. (Ref. 118) presented some interesting results concerning saturated water flow in tubes with nonuniform heating along their length. They showed that the effect of nonuniform heating on DNB (CHF) was similar to that of Ornatskiy's team (CHF for nonuniform heating is less than CHF for uniform heating); however, as the region over which the nonuniform heating occurred became very small (i.e., a spike heat flux), the CHF for the nonuniform heating case approached and surpassed that for a uniformly heated tube.

Aladyev, et al. (Ref. 44) demonstrated that nonuniform heating around the perimeter of a tube with subcooled or saturated water flow can effect the level of CHF. They showed that the CHF generally occurs where the heat flux is the highest and increases with increasing unevenness of the heating (characterized by the ratio of the maximum to the average heat flux). It was also shown that the effect of nonuniform heating around the perimeter decreased with increasing diameter. Leontiev, et al. (Ref. 119) found, for subcooled water flowing in horizontal tubes with nonuniform circumferential heating (the ratio of maximum heat flux to average heat flux was 1.5), that the local values of CHF were somewhat higher than those of uniformly heated tubes. However, when they averaged the heat flux at the station where the critical point occurred, the averaged CHF for the nonuniform heating case was much lower than that for the uniform heating case (at subcooled conditions), and the two coincided at high quality conditions.

Transient Operation-

The effects of transient power (heat flux), flow rate, and pressure operation on heat transfer have been studied in previous experiments primarily associated with petrochemical and nuclear power plant safety (e.g., Ref. 120). A major portion of these experiments involved the study of transient heat flux effects. Ragheb, et al. (Ref. 121), showed that the wall superheat was larger for transient boiling (found by quenching a high thermal capacity tube with subcooled, low pressure water flow) than steady-state boiling at comparable conditions (Fig. 32). Cheng, et al. (Ref. 122) had earlier found little difference in wall superheat for transient or steady-state heating with the same apparatus, though a limitation in the steady-state heating capability prevented data comparison at heat flux levels approaching the CHF. Kataoka, et al. (Ref. 123) found a similar effect on wall superheat as Ragheb's team for an exponentially increasing heat input to a platinum wire in subcooled forced water flow.

Several interesting results have been found during studies of the effect of transient heat flux on CHF. Borishanskiy and Fokin (Ref. 62) and later Tolubinskiy, et al. (Ref. 124) found that the CHF were equal for transient and steady-state heating in pool and forced convection boiling of water with and without subcooling. However, when organic coolants[8] were used it was found that the CHF for steady-state heating was much larger than the CHF for transient heating (Tolubinskiy's team found this difference to be as large as a factor of 3). Tolubinskiy's team theorized that the transient operation is viscosity dependent and pointed out that the organics have higher viscosities than that of water, thereby contributing to the difference in CHF for the transient and steady-state heating of organic fluids. Roemer (Ref. 125) found no effect of transient heating (time constant of the applied power was approximately 0.1 sec) on CHF for water flow normal to cylindrical test elements as long as the test surfaces were aged. This result was found to be independent of the test element wall thickness and the speed of the power transient. The CHF for unaged specimens were as much as 25 percent lower than the CHF at comparable steady-state conditions. Other researchers have found results conflicting with those observed by Borishanskiy's team and Tolubinskiy's team for water. As seen in Fig. 32, the data of Ragheb, et al. (Ref. 121) show the CHF for transient conditions to be higher than that for steady-state conditions. Kataoka, et al. (Ref. 123) also found the CHF for transient heating to be greater than that for steady-state heating, and the difference between the two CHF varied approximately as the -0.6 power of the period of the exponentially increasing heat flux (Fig. 33).

A fewer number of experiments have been performed to evaluate the effects of transient flow rate or pressure operation on boiling heat transfer. Celata, et al. (Ref. 126), using subcooled Refrigerant-12 as a coolant, found that the difference between the CHF for a flow rate transient and the CHF for a steady-state flow rate became significant when the time to reach half of the initial flow became less than approximately twice the flow rate transit time. The CHF for the transient flow was typically less than the comparable steady-state CHF in these cases. Celata, et al. (Ref. 127) later evaluated the combined effect of transient flow rate and thermal power on CHF using flowing subcooled R-12. His team found that the CHF for the transient cases were approximately equal to those for steady-state conditions when the time to reach half flow was greater than the transit time. When the time to reach half flow dropped below the transit time, the CHF for the transient case became greater than the corresponding steady-state CHF. These trends appeared to be insensitive to the level of pressure or mass flux at which the runs were made. Weisman, et al. (Ref. 128), using depressurization of saturated water to produce boiling in a nonflowing apparatus, found that the wall superheat for the pressure transients was significantly higher (as high as a factor of two) than a slow pressure change, though the data is probably applicable only near boiling incipience. Celata, et al. (Ref. 129) used flowing subcooled R-12 to demonstrate that pressure transients tended to reduce the CHF (higher depressurization rates led to more marked crisis conditions) from that exhibited during comparable steady-state conditions. Finally, Celata, et al. (Ref. 120) evaluated the combined effect of transient pressure, power and/or flow rate on CHF though no steady-state data were obtained for comparison. Celata's team did show the inadequacy of using the available steady-state CHF correlations in predicting transient power/flow rate/pressure situations.

[8] Borishanskiy's team used ethyl alcohol while Tolubinskiy's team used ethanol and acetone.

Flow Instabilities-

Thermal-hydraulic flow instabilities are directly related to the flow system design, and can seriously affect the value of CHF and the integrity of the flow system (Ref. 17). Podowski (Ref. 130) identified several instabilities that can appear in two-phase flow and pointed out that these instabilities can be compounded in some cases. The classes of instabilities include excursive (Ledinegg), flow regime relaxation, nucleation, density-wave oscillation, pressure drop oscillation, acoustic, and condensation-induced instabilities. Numerous authors (Refs. 17, 130-133) have provided excellent reviews of these instabilities and their effects, therefore, a rigorous treatment is excluded here. In general, flow instabilities have a detrimental effect on boiling heat transfer, primarily a reduction in the CHF (Ref. 14).

According to Podowski (Ref. 130) one of the most predominate of the instabilities are those classified under density-wave oscillation. They include flow loop instabilities, parallel-channel instabilities, and channel-to-channel instabilities. Flow loops consisting of sections with liquid flow combined with sections having liquid/vapor flow will have areas with different speeds of propagation of perturbations. In such a system a perturbation can induce velocity and pressure oscillations which may diverge or reach a self-sustained periodic mode. In the latter case the oscillations can exceed the thermal limits of the system thereby reaching a premature CHF (e.g., Silvestri, Ref. 134, found that the CHF was reduced by as much as 40 percent when flow loop instabilities were present in coexisting water vapor/liquid flow). Premature CHF have also been demonstrated in parallel channels (see Veziroglu and Lee, Ref. 135) that share a common inlet manifold or have a bypass channel. In configurations where the channels have similar operating conditions, the phase of oscillations may be opposite depending on the number of channels (Ref. 130). Such oscillations are called channel-to-channel instabilities. In many cases, the manifolded cooling approach is used in high enthalpy arc facilities and may be prone to these types of instabilities when two-phase flow is present.

Density-wave oscillations frequently combine with other phenomena, such as pressure-drop oscillations, to create additional instabilities. For instance, Lowdermilk, et al. (Ref. 98) and later Aladyev, et al. (Ref. 44) showed that pulsations in flowing subcooled and saturated water due to an upstream compressible volume reduced the CHF (from that of a stable system with no pulsations) as much as 80 percent. Lowdermilk's team found that the flow instability could be removed by throttling the flow just upstream of the test section (Fig. 34). In addition, Gambill, et al. (Ref. 113) suggested that stability is improved as the test section pressure drop becomes a smaller fraction of the total system pressure drop, and restricted flow upstream of the test section for most of their tests[9]. It appears that the stability of a cooling apparatus which involves flow boiling can be improved by avoiding compressible volumes in the flow loop (or throttling the flow between the

[9] Gambill's team restricted the ratio of test section pressure drop to the total system pressure drop to values between 0.045 and 0.40.

compressible volumes in the flow loop (or throttling the flow between the compressible volume and the test section) and forcing the test section pressure drop to be a small portion of the total pressure drop of the system. However, as pointed out by Boyd (Ref. 17), in cases where a compressible volume is in the test section, or in the case of very long test sections, no amount of throttling can eliminate or reduce the instability.

Heat Application/Orientation-

Though not typically associated with an actual cooling configuration, the method of heat application when resistive (Joule) heating is used in an experimental simulation can be important. Ellion (Ref. 136) was one of the first to note an effect of alternating current (a-c) heating on an experimental boiling apparatus. Ellion found that a 60 cycle/sec a-c power supply caused a 120 cycle/sec growth and collapse of the bubbles in flowing subcooled water in an annulus. The problem was rectified by switching to direct current (d-c) power with the amplitude of the voltage ripple less than one percent over the full operational range. Gambill, et al. (Ref. 113) suspected a similar problem when they used a-c power to electrically heat a tube with internally flowing subcooled water. Gambill's team calculated that the a-c heating reduced the CHF as much as 16 percent over that for d-c heating. Tippets (Ref. 28), who used a-c heating of stainless steel ribbons to study subcooled and saturated water flow boiling, found that the CHF for 0.006-in. thick heater ribbons averaged 20 percent lower than those for 0.010-in. thick ribbons and in a few cases the difference was substantially larger. Tippets theorized that the reduction was due in part to the lesser thermal time constant of the thinner ribbons coupled with the a-c heating.

Leung, et al. (Ref. 43) found little effect (less than 6 percent difference) of direct and indirect heating of the interior wall of an annulus with flowing water at subcooled inlet conditions. As mentioned previously, Collier (Ref. 7) points out that the outer wall in a concentric annulus system has little or no effect on the CHF for the inner surface. However, Tong (Ref. 10) suggests that the presence of an unheated wall in the proximity of a critical point can adversely affect the cooling effectiveness (termed the "unheated wall effect"). Tolubinsky, et al. (Ref. 137) found, for subcooled and saturated water flowing in annuli, that higher CHF (as much as 30 percent) existed on a heated inner wall than on a heated outer wall at comparable conditions. Yücel and Kakaç (Ref. 138) found that the heater orientation had an effect on the CHF in a rectangular channel with flowing subcooled water. Their data indicated that higher CHF are achieved with the heated surface (along one side of the rectangular channel) facing up rather than facing down. They attributed the difference, though, not to the presence of an unheated wall, but to buoyancy forces moving the vapor away from the lower surface resulting in a lower surface temperature and higher CHF (as opposed to a higher surface temperature and lower CHF on the downward facing heater).

Miscellaneous Effects-

Several other parameters have been shown to have an effect on the cooling process albeit only a small number of experiments have been performed that demonstrate their effect. Tong (Ref. 10) suggested that the local enthalpy can impair the CHF, primarily when there is significant vapor voidage near the heated wall. Such conditions could exist for subcooled[10] or saturated flow even at high velocity. The local enthalpy near the wall even at upstream locations can be excessively high and cause burnout when significant vapor exist in the bubble layer near the wall.

Johnson (Ref. 139) pointed out that the presence of an electric field has some effect in the pure convection region, little effect in the nucleate boiling region, and a significant effect at CHF and beyond. Markels and Durfee (Ref. 140) applied a 3000-v a-c voltage across the gap of an annulus with the inner wall heated and flowing subcooled water. They found that the CHF were augmented as much as 40 percent with the applied electric field as opposed to no field present. No effect of the applied voltage was found where flow instability burnouts occurred.

Another heat flux augmentation method which has been demonstrated involves the use of flow and ultrasonic vibration. Although the use of vibration has not resulted in an increase in CHF for forced flow configurations (Refs. 7 and 141), an enhancement of the heat transfer (increase in the heat transfer coefficient) in the single-phase forced convection region and boiling onset has been demonstrated (see Bergles, Ref. 141).

Boyd (Ref. 1) identified two other heat flux enhancement techniques which should be included here for completeness. As Boyd noted, Inoue and Bankoff (Ref. 142) found that the destabilization of film boiling (initially subcooled) by the introduction of a pressure wave increased the transient heat transfer up to 20 times the steady-state values. Boyd also noted that Weede and Dhir (Ref. 143) showed that the CHF can be enhanced and its location controlled using tangential injection of a fluid into the mainstream. This technique probably takes advantage of the induced swirl and turbulence whose benefits were discussed previously.

3.2 SUMMARY OF PARAMETRIC TRENDS AND IMPLICATIONS

As suggested previously, velocity and subcooling have a primary contribution on cooling effectiveness, and it appears that a highly subcooled, forced convection cooling system is most attractive for high enthalpy arc facility component cooling. Figure 35 summarizes the effect of a majority of the parameters and enhancement techniques previously discussed for a subcooled, forced convection cooling environment. Those parameters and techniques that tend to decrease heat transfer

[10] Fiori and Bergles (Ref. 20) found, contrary to prior beliefs, that large vapor voidages can exist even in subcooled flow. They noted that instantaneous void fractions larger than 50 percent were possible at high heat flux ($1x10^6$ to $5x10^6$ Btu/ft^2), high subcooling (50 to 100°F), and low pressure (less than 100 psia).

or increase wall temperature in the pure convection (nonboiling) and nucleate boiling regions, respectively, are not necessarily detrimental to the cooling process unless the wall temperature increase approaches the melt temperature of the material or contributes to plasticity to the point of structural failure. Probably the most important contributions to cooling effectiveness and, hence, component survival, are those parameters and enhancement techniques which tend to increase the CHF. A number of parameters, such as pressure, can have either a beneficial or a detrimental effect on CHF and must be optimized for the conditions of interest. However, system limitations and configuration requirements may limit the degree of optimization that can be achieved.

The implications of requiring a subcooled, forced convection cooling configuration are numerous and not generally conducive to experimental evaluation. As previously discussed, high velocity, high subcooling, and high pressure tend to decrease bubble diameter and lifespan making their detection and observation difficult (Ref. 144). In addition to data shown in Fig. 8, Gunther (Ref. 27) found that the bubble size and lifetime also decrease with increasing heat flux (Fig. 36). Bubble sizes on the order of 100 μm (0.004 in.) and lifetimes on the order of 0.1 msec would not be unreasonable for high subcooling, high velocity, high heat flux conditions. Another disturbing trend is the rapidity with which a cooling system transitions from nucleate boiling to burnout at these conditions. Hughes (Ref. 21) noted that the speed and intensity of the transition to film boiling (i.e., burnout) was observed to increase with increasing velocity and subcooling. Knowles (Ref. 145) noted rapid transitions in an annulus with subcooled water flow that in some cases caused destruction of not only the heated inner tube but the annulus assembly as well. Cavitation also becomes important if high coolant velocity is desired but the apparatus is limited to lower pressure operation. Inadequate cooling (causing premature burnout) and/or structural damage can occur at or near the point in the apparatus where the cavitation takes place. Finally, the requirements of high velocity, high pressure, and high heat flux capability significantly increase the complexity of the cooling configuration, whether it is used in an experimental apparatus or the actual cooled component assembly. The increased complexity also increases the difficulty of incorporating instrumentation to monitor or determine operating conditions.

4.0 THEORETICAL AND CORRELATIONAL PREDICTION OF THE BOILING CURVE

Numerous theoretical models and correlational approaches have been proposed for various regimes and mechanisms in convective heat transfer, including boiling, and fluid dynamics over the past several years. As mentioned previously, cooling at low heat flux by pure convection heat transfer is fairly well understood; however, even with the considerable attention that boiling heat transfer has received over the past 50 years, our understanding of the mechanisms involved in boiling necessary for successful modeling of the phenomena is considerably lacking. It should be kept in mind that, although the following discussion is centered around forced convection and flow boiling heat transfer up to and including CHF, extensive work has also been performed in natural convection, pool boiling, and film boiling.

4.1 PURE FORCED CONVECTION

Although complex computational fluid dynamics techniques are available (see, for instance, Chapters 4 and 6 of White - Ref. 146), analytical solutions and experimental correlations (for more complex flow geometries) have been derived which allow for quick and accurate prediction of heat transfer and flow conditions in various cooling configurations. Several familiar turbulent flow heat transfer correlations based on the Reynolds analogy were developed in the 1930's for tube flow, most notably those of Dittus and Boelter (for heating liquids),

$$Nu_D = 0.0265\left(Re_D\right)^{0.8}\left(Pr\right)^{0.4} \tag{1}$$

Colburn (for heating or cooling in tubes),

$$Nu_D = 0.023\left(Re_D\right)^{0.8}\left(Pr\right)^{1/3} \tag{2}$$

and Sieder and Tate (for heating or cooling in tubes),

$$Nu_D = 0.027\left(Re_D\right)^{0.8}\left(Pr\right)^{0.8}\left(\mu/\mu_w\right)^{0.14} \tag{3}$$

where the fluid properties in the first two expressions are evaluated at a mean temperature $(T_b + T_w)/2$, and at the local bulk fluid temperature in the Sieder-Tate expression (see Lienhard, Ref. 3). The first two correlations assume a low temperature difference between the bulk fluid and the tube surface. Sieder and Tate modified the Colburn correlation to account for a much larger temperature difference. Improvements to these correlations were later made by Carpenter, et al. (Ref. 147), who modified the Sieder-Tate correlation for use with annuli,

$$Nu_{D_e} = 0.023\left(Re_{D_e}\right)^{0.8}\left(Pr\right)^{1/3}\left(\mu/\mu_w\right)^{0.14} \tag{4}$$

where D_e (the equivalent or hydraulic diameter) is used for the characteristic diameter, and Kays and London (Ref. 148), who presented Nusselt no. and friction factor information in graphical form for various channel geometries (including tubes and annuli), flow field development, and heating arrangements. Though these correlations gave reasonable accuracy for most situations, as pointed out by Lienhard (Ref. 3), much greater accuracy over a broader range of conditions can be achieved with the more modern correlation derived by Petukhov (Ref. 149) for tube flow,

$$Nu_D = \left(f/8\right) Re_D Pr / \left\{ 1.07 + 12.7\ SQRT\left(f/8\right)\left(Pr^{2/3} - 1\right)\right\} \left(\mu_b/\mu_w\right)^n \qquad (5)$$

where,

$$\mu_b/\mu_w \textit{ is between } 0 \textit{ and } 40$$

$$Re_D \textit{ is between } 10^4 \textit{ and } 5 \times 10^6$$

$$Pr \textit{ is between } 0.5 \textit{ and } 200 \textit{ for } 6\% \textit{ accuracy}$$

$$Pr \textit{ is between } 200 \textit{ and } 2000 \textit{ for } 10\% \textit{ accuracy}$$

$$n = 0.11 \textit{ for } T_w > T_b$$

$$n = 0.25 \textit{ for } T_w < T_b$$

$$n = 0 \textit{ for gases}$$

and where,

$$f = 1/\left(1.82 \log_{10} Re_D - 1.64\right)^2 \quad \textit{for smooth pipes}$$

or from Moody chart for smooth or rough pipes

Sleicher and Rouse (Ref. 150) recommended a simpler expression with comparable accuracy for tube flow,

$$Nu_D = 5 + 0.015\left(Re_D\right)^a\left(Pr\right)^b \qquad (6)$$

where,

$$a = 0.88 - 0.24/\left(4 + Pr\right)$$

$$b = \left(1/3\right) + 0.5\, e^{(-0.6\, Pr)}$$

and

$$Pr \textit{ is between } 0.1 \textit{ and } 10^5$$

$$Re_D \textit{ is between } 10^4 \textit{ and } 10^6$$

The properties in Re_D are evaluated at the local film temperature, $T_f = (T_b + T_w)/2$, and the properties in Pr are evaluated at the local wall temperature.

Kays and Leung (Ref. 151), solved the turbulent-flow energy equation for constant heat flux in annuli over wide ranges of Reynolds number, Prandtl number, and annulus radius ratio. Table 1 in Ref. 151 or Tables 13-3 to 13-5 in Ref. 5 present the results of their computations.

For fully developed laminar flow in a circular tube with uniform heat flux, the Nusselt number and, hence, the heat transfer coefficient is constant, independent of Reynolds number and Prandtl number,

$$Nu_D = 4.36 \qquad \textit{constant heat flux}, Pr > 0.6 \tag{7}$$

Treatment of heat transfer in the entry region where velocity and temperature profiles are not fully developed, is discussed in Chapter 8 of Ref. 152.

Pressure drop is also of interest in a cooling process and the pressure gradient has even been shown to have an effect on local heat transfer (see Chapter 6 of Ref. 146). Solution of the equations of motion (using semiempirical shear correlations for the turbulent case where random fluctuations are important) provides a means of determining the pressure drop in a flow cooling configuration as outlined by White (Ref. 153). Friction factors as a function of Reynolds number for flow in smooth of rough circular tubes, annuli, or noncircular ducts are provided by the Moody chart or expressions such as those presented in Chapter 6 of Ref. 153.

4.2 TRANSITION TO FLOW BOILING, FLOW BOILING, AND CHF: THEORETICAL MODELS

The theoretical determination of heat transfer, pressure drop, and other flow characteristics once a cooling process has transitioned into boiling is one of the greatest challenges facing researchers in two-phase flow. Three currently accepted, two-phase flow or "mixture" models have been developed based on the more dominant flow regimes in two-phase flow. The *drift flux* model, which is based on a molecular diffusion model, allows for motion of one phase relative to the other and the two are coupled through the use of correlations accounting for interaction effects. The correlations are difficult to obtain because they require a knowledge of the average effects of one phase moving relative to the other (see, for example, Ref. 154). Another theoretical model, the *two-fluid* model based on the application of the Navier-Stokes equations, treats the phases separately and requires matching conditions at both solid boundaries and liquid-vapor interfaces. Difficulty is encountered in obtaining local and instantaneous mass, momentum, and energy balance equations at the interfaces (see, for example, Ref. 155). The drift flux model is typically more applicable to flows where the one phase is dispersed throughout the other. The two-fluid model is generally used for flows where the two phases are separated into layers, but has been accepted for use with dispersed flows. If the motion of the two phases can be assumed to be equivalent (i.e., no slip condition), a simplified model, the *homogeneous* model, that treats the mixture as a single-phase fluid, may be used. The transport properties required for the model are difficult to determine due to the lack of existing relationships. Though extensive progress in theoretical modeling has been achieved in recent years, its application has been limited because of the difficulties described above. A more complete treatment of these models, especially the two-fluid model, is included in Jones, Ref. 156, and Drew, Ref. 157.

4.3 TRANSITION TO FLOW BOILING: CORRELATIONAL APPROACHES

Early studies in flow boiling suggested that boiling incipience occurred at the intersection of the pure convection and fully developed nucleate boiling curves, although experimental data indicated a transition region (McAdams, et al., Ref. 24). Several investigators later derived heat transfer expressions for the transition from pure convection to fully developed nucleate boiling (termed partial nucleate boiling). Kutateladze (Ref. 158) and, later, Forster and Greif (Ref. 56) suggested correlations based on the pure convection heat flux at the incipience of boiling and the corresponding value of heat flux in pool boiling. Bergles and Rohsenow (Ref. 159), however, later showed that the transition region between pure convection and nucleate boiling cannot be based on results from saturated pool boiling, and recommended the following relation based on an extension of the fully developed nucleate boiling curve,

$$q''/q''_{conv} = \left[1 + \left\{\left(q''_B/q''_{conv}\right)\left(1 - q''_{Bi}/q''_B\right)\right\}^2\right]^{1/2} \tag{8}$$

where q''_B is calculated from a fully developed nucleate boiling correlation at various wall temperatures, and q''_{Bi} is determined from an extension of the fully developed nucleate boiling curve to the temperature corresponding to the boiling incipient point (see Fig. 37). In addition, Bergles and Rohsenow developed a relation for heat flux at subcooled boiling incipience by solving a bubble growth equation graphically to obtain,

$$q''_{LB} = 15.60\, p^{1.156}\left(T_w - T_{sat}\right)^{2.3/p^{0.0234}} \tag{9}$$

4.4 FLOW BOILING: CORRELATIONAL APPROACHES

The team of McAdams, et al. (Ref. 24) was one of the earliest boiling research teams to proposed an empirical correlation for fully developed nucleate boiling in subcooled forced flow. They found for flow in an annulus (heated stainless steel inner wall) that the heat flux during boiling was independent of coolant velocity (1 to 36 fps), degree of subcooling (20 to 150°F), pressure (30 to 90 psia), and equivalent diameter (0.17 to 0.48 in.). They correlated the heat flux data with the expression,

$$q'' = C'\Delta T_{sat}^{\;3.86} \tag{10}$$

where the value of C' decreased from 0.19 to 0.074 as the concentration of dissolved gas decreased from 0.30 to 0.06 ml of air at standard conditions per liter of water. Jens and Lottes (Ref. 85) analyzed high-pressure, high heat flux, flow boiling data which were obtained by researchers at UCLA, Purdue, and MIT, and derived the following empirical correlation,

$$q'' = \left(0.527\, e^{p/900} \Delta T_{sat}\right)^4 \tag{11}$$

where p is in psf. Jens and Lottes found that the wall temperature during boiling was independent of the flow velocity, gas content, and degree of subcooling and was dependent only on heat flux and water pressure. All of the data were obtained for flow inside round or square tubes made of stainless steel or nickel.

Gilmour (Ref. 160) derived a correlation based on dimensionless groups rather than temperature difference:

$$\left(h/c_l G\right)\left(c_l \mu_l / k_l\right)^{0.66}\left(\rho_l \sigma / p^2\right)^{0.425} = 0.001/\left(DG/\mu_l\right)^{0.3} \tag{12}$$

where,

$$G = \left(\dot{m}_v / A\right)\left(\rho_l / \rho_v\right)$$

and where p is in psf. Although Gilmour substantiated the correlation with data from various experiments including several in which water was used, it was noted that surface material had a significant effect that was not accounted for in the correlation and that data from experiments involving boiling from wires did not agree well with the correlation.

Later, Levy (Ref. 161) derived a semiempirical correlation based on the Forster and Zuber (Ref. 162) bubble growth model and applicable to nucleate boiling with flow as well as pool boiling, arguing that the fluid velocity normally used exerts a minor effect in comparison with the local turbulence produced by the bubble motions. For subcooled liquids, Levy proposed the correlation,

$$q'' = \left[k_l c_l \rho_l^2 / T_s \left(\rho_l - \rho_v\right)\right]\left[1/B_L\right]\left[\left\{i_{fg} + c_l\left(T_s - T_b\right)\right\} / i_{fg}\right]\left(\Delta T_{sat}\right)^3 \tag{13}$$

where T_s and T_b are in °R, σ in Btu/ft^2, and B_L is an empirical constant which is a function of $\rho_v i_{fg}$ as shown in Fig. 38. Levy found that the above correlation agreed reasonably well with data from McAdams, et al., and the UCLA data used by Jens and Lottes discussed previously.

About the same time Levy derived his correlation, Forster and Greif (Ref. 56) theorized that the primary boiling heat transfer mechanism is the exchange between the vapor and liquid, and that microconvection in the liquid sublayer beneath the bubbles, bubbles acting as artificial roughness, and latent heat transport by the bubbles contribute very little. Based on this argument, they used a

Reynold's analogy to derive the following expression for general boiling of various liquids including water:

$$q'' = K_{sf}\left[\alpha c_l \rho_l T_s / J i_{fg} \rho_v \sigma^{1/2}\right]\left[c_l T_s \sigma^{1/2} / J\left(i_{fg}\rho_v\right)^2\right]^{1/4}\left[\rho_l/\mu_l\right]^{5/8}\left[\mu_l c_l / k_l\right]^{1/3}\Delta p^2 \qquad (14)$$

where Δp is the pressure difference in psf corresponding to the superheat, ΔT_{sat}. Reasonable agreement was achieved when the correlation was compared to experimental data at pressures up to 50 atm with K_{sf} equal to 0.0012. In the discussion in Ref. 56 the authors noted that the constant does change for other data.

Shah (Ref. 163) presented a correlation for the generalized prediction of heat transfer during subcooled boiling in annuli based on earlier work involving tubes (Ref. 164). His correlation is,

$$q'' = h_l\left(T_w - T_b\right) + h_l\left(\psi_o - 1\right)^3\left(T_w - T_{sat}\right) \qquad (15)$$

where,

$$\psi_o = 1 + 46\,B_o^{0.5}\ for\, B_o < 0.3 \times 10^{-4}$$

$$\psi_o = 230\,B_o^{0.5}\ for\, B_o > 0.3 \times 10^{-4}$$

$$B_o = q''/G\, i_{fg}$$

$$h_l = nonboiling\ heat\ transfer\ coefficient$$

Shah found that the expression correlated 450 representative data points with a mean deviation of 8.9 percent. The range of test conditions were: reduced pressure from 0.009 to 0.89, subcooling from 2 to 109 °C (36 to 228 °F), mass flux from 278 to 7774 kg/m^2 sec (57 to 1592 lbm/ft^2 sec), inner tube diameter from 4.5 to 42.2 mm (0.177 to 1.66 in.), and annular gap from 1 to 6.6 mm (0.039 to 0.26 in.). Various fluids, including water, were evaluated and heating was from the inner, outer, or both walls of the annuli.

Probably the most widely accepted and verified flow boiling correlation is based on Rohsenow's (Ref. 165) pool boiling correlation (a Nusselt number derived from the fluid Prandtl number and a bubble Reynolds number) and proposed by Rohsenow (Ref. 166) in 1952. The heat-transfer rate associated with forced-convection boiling was found to be accurately predicted by addition (or superposition) of the pool-boiling effect with the pure forced-convection effect discussed above, i.e.,

$$q'' = q''_{forced\ convection} + q''_{pool\ boiling} \qquad (16)$$

where $q''_{\text{forced convection}}$ is determined from the expressions presented in the previous discussion of pure forced-convection correlations and $q''_{\text{pool boiling}}$ is determined from,

$$c_l\left(T_w - T_{sat}\right)/ i_{fg}Pr^s = C_{sf}\left[\left\{q''/\mu_l i_{fg}\right\}\left\{g_c\sigma/g\left(\rho_l - \rho_v\right)\right\}^{1/2}\right]^r \tag{17}$$

where the properties are evaluated at T_{sat} (except as noted), r equals 0.33, s equals 1 for water cooling and 1.7 for other fluids, and C_{sf} equals 0.006 for nickel and brass surfaces and 0.013 for copper and platinum surfaces[11]. Vachon, et al. (Ref. 167) later presented new values of r and C_{sf} which account for surface preparation technique as well as liquid-surface combination. However, it should be noted that their analysis applies only to the linear portion of the nucleate boiling curve, i.e, that portion which excludes partial nucleate boiling and approaching CHF. In addition, the effects of subcooling and pressure have not been fully investigated and may effect the constants presented by Vachon's team. Tong (Ref. 10) pointed out that the superposition approach may be questionable in light of the experiments of Bergles and Rohsenow (Ref. 159) which demonstrated the differences in the fluid mechanics of flow boiling and pool boiling. Bergles and Rohsenow recommended a less restrictive construction approach for predicting forced-convection boiling when the net vapor generation is relatively small, which was presented previously in Eqn. 8. Bjorge, et al. (Ref. 168) and Stephan and Auracher (Ref. 169) later presented variations of the superposition approach. For additional approaches, Guglielmini, et al. (Ref. 170) provided a survey of flow boiling correlations as of 1980, including boiling incipience and partial nucleate boiling expressions.

Large vapor void fractions have been observed in low-pressure, subcooled flow, possibly greater than 20 percent on a time-averaged basis and exceeding 50 percent on a local instantaneous basis (Ref. 20). Such void fractions contribute to the accelerative and static head components of the pressure gradient along the heated channel. Flow boiling experiments where pressure drop characteristics were evaluated (e.g., Refs. 22 and 171) generally have shown that as heat flux increases the pressure drop decreases until boiling begins, after which the pressure drop increases (Fig. 39). Boyd (Ref. 172), however, found pressure drops for high pressure, moderately subcooled water boiling to be slightly less than single-phase pressure drops. Boyd noted that CHF occurred before substantial pressure drops occurred. Bergles and Dormer (Ref. 171), who performed experiments in subcooled water flow boiling at pressures below 100 psia, noted that little success had been reported in the general correlation of pressure drop and presented graphical correlations for the pressure drops encountered in their experiments. About the same time, Staub and Walmet (Ref. 173) identified the two regions before and after the point of significant vapor generation (SNVG) where the effects of void fraction on pressure

11 The Rohsenow correlation is valid only for clean, relatively smooth surfaces. Rohsenow (Ref. 165) found that s varied erratically between 0.8 and 2.0 when the surface was dirty.

drop are different in subcooled flow boiling. Recently, Hoffman and Kline (Ref. 174) and Jia and Schrock (Ref. 175) have developed correlations for the prediction of void fraction and the subsequent gravity, acceleration, and friction pressure drop before and after SNVG.

4.5 CHF CORRELATIONAL APPROACHES

No theory of subcooled burnout has yet been created for flow boiling (Ref. 33). Complexity of the boiling mechanisms at the critical point, their unsteady nature, and the small scale of the heat transfer processes have hindered the development of theoretical models and understanding of the phenomena from experimental studies. Moreover, the cooling flow structure and, hence, the heat transfer processes vary depending on the heating condition, fluid properties, and flow/surface geometries as indicated in the previous discussion of parametric effects. This has led to an excessive number of correlations derived for the prediction of CHF in flow boiling. Most of the CHF correlations, not unlike those for fully developed nucleate boiling, were developed either by the analytical (mechanism-based) approach, dimensional analysis/similitude-based, or optimization of test data (empirical-based).

4.5.1 Mechanism-based CHF Correlations

In an attempt to derive analytical relationships among the macroscopic heat transfer and flow field characteristics in a boiling environment, several models describing the mechanisms on a microscopic scale have been proposed. The mechanisms appear to vary with the degree of subcooling, and several current approaches divide the subcooled boiling regions into three areas: (1) high subcooling and mass velocity, (2) moderate subcooling, and (3) low subcooling, velocity, and pressure (Ref. 8). At high subcooling and mass velocity, it has been proposed that local wall overheat occurs when high heat flux leads to dryout and a resulting hot spot beneath a bubble; however, there is little experimental support and no correlations have been developed based on this mechanism (Ref. 8). At moderate subcoolings, it has been proposed that the liquid flow stagnates near the heated wall when the boundary layer separates. At high heat flux the stagnant liquid evaporates resulting in vapor blanketing near the wall and subsequent burnout. Tong (Ref. 176) presented a correlation based on this mechanism which agreed reasonably well with data from subcooled, high-pressure water flow in tubes, annuli, and rod bundles. Purcupile and Gouse (Ref. 177) presented a similarly based correlation in nondimensional form. Other studies associated with this proposed mechanism include the work of Kutateladze and Leont'ev (Ref. 178) and Hancox and Nicoll (Ref. 179). At low subcooling, velocity, and pressure, Fiori and Bergles (Ref. 20) suggested that dry-out of the microlayer follows when inadequate quenching of the surface by adjacent liquid occurs after the passage of a bubble. Subsequent burnout occurs when high surface temperatures prevent rewetting of the surface. Because of difficulties in defining the flow structure, no correlation could be obtained by the authors. Lee and Mudawwar (Refs. 180 and 181) proposed a similar mechanism by suggesting that CHF occurs as a result of the Helmholtz instability at the microlayer-vapor interface, leading to dryout of the microlayer. They demonstrated that their correlation agreed with numerous subcooled (over a wide range of subcoolings) water data points with a mean deviation of approximately 11 percent. Katto (Refs. 182 and 183) presented a correlation based on this same sublayer dryout mechanism and suggested that CHF of subcooled flow boiling has strong similarities to pool boiling CHF. Earlier, Katto first presented generalized correlations for CHF in subcooled and saturated water flow in tubes (Refs. 184 and 185) and annuli (Ref. 186) based on the existence of four characteristic regimes of CHF and the following dimensionless groups,

$$q''_{CHF} / \left\{ G i_{fg} \left[1 + K \left(\Delta i_l / i_{fg} \right) \right] \right\} = f\left(\rho_v / \rho_l, \sigma \rho_l / G^2 L, L/D \right) \tag{18}$$

where K is determined analytically to account for inlet subcooling. A reasonable correspondence between the correlations and experimental data was found, however, later studies (e.g., Ref. 187) identified problems with applying the correlations to the various regimes. In addition, difficulty was encountered in deriving an expression for CHF at very high mass velocity and pressure conditions. Nishikawa, et al. (Ref. 188) later correlated high pressure tube data at various mass flows with Katto's proposed dimensionless groups using the boiling length as the characteristic length. Their proposed expressions correlated approximately 150 data points to within ± 10 percent.

Other proposed mechanisms of general subcooled flow-boiling CHF include liquid flow blockage models and vapor removal limit/bubble crowding models (Ref. 182). The liquid flow blockage models assume that the flow of liquid normal to the heated surface is blocked by the flow of vapor along the surface. The mechanisms proposed by Bergel'son (Ref. 189) and Smogalev (Ref. 190) involve some form of liquid flow blockage. Griffith (Ref. 191) presented a dimensionless correlation based on a critical packing of bubbles on the surface. Griffith proposed that a saturated pool boiling bubble growth term be multiplied by a correction factor which included velocity and subcooling effects. Approximately 94 percent of his data were correlated by the expression to within ± 33 percent. Hughes (Ref. 21) suggested that CHF occurred when a maximum number of nucleation sites available for a surface was exceeded (i.e., bubble crowding). For a straight surface he proposed the correlation:

$$\begin{aligned} q''_{CHF} = {} & \left[2 d_s \rho_v \left(c_l \Delta T_{sub} + 2 i_{fg} \right) \right] / 3K \left\{ 4/9 \pi\alpha \left(\rho_l \Delta T_{bulk}\, d_s / \rho_l \Delta T_{sat} \right)^2 + \right. \\ & \left. \pi\alpha/3 \left(\rho_v i_{fg} d_s / k_l \Delta T_{sat} \right)^2 + \left(1/B \right) ln \left[\left(U_o + B d_s \right) / U_o \right] \right\} \end{aligned} \tag{19}$$

where,

$$d_s = 6.12\, W D_h / Re^{7/8} \text{ (W = 30 } in\, Ref.\ 21\text{)}$$

$$B = (12/11)(C_D \mu_l / \rho_l d^2)$$

$$U_o = (3/4\pi a)(k_l^{\,2} \Delta T_{sat}^{\,2} / \rho_v^{\,2} i_{fg}^{\,2} d)$$

and ρ is in lbm/in3, k is in Btu/in sec °F, μ is in lbm/in sec, D_h and d are in inches, and a is in in2/sec. The term K was determined experimentally as,

$$K = \beta_{fs} V_m^{\,1/2} / \Delta T_{sub} \quad (\beta_{fs} \textit{ is approximately equal to } 7.94)$$

Although not given by Hughes, the drag coefficient for a bubble is approximated by

$$C_D = 32 a \mu_l / \rho_l V_m d^2 \quad (\textit{see Refs. } 146 \textit{ and } 153)$$

and, for a straight test section, the bubble diameter for the preceding equations is approximated by

$$d = d_s$$

Hughes also presented somewhat cumbersome algebraic expressions for CHF on convex and concave surfaces (see Ref. 21). Hebel, et al. (Ref. 192) similarly suggested that CHF occurs when a limiting vapor removal rate is reached. Other proposed mechanisms included in this group are those suggested by Yagov and Puzin (Ref. 193) and later, Weisman and Ileslamlou (Ref. 194). Weisman and Ileslamlou (see also Yang and Weisman - Ref. 195), by extending the earlier work reported in Ref. 196 based on limited turbulent interchange at the outer edge of a bubbly layer near a heated surface, proposed the following CHF correlation for use in highly subcooled flow:

$$q''_{CHF} = G \Psi i_b \left[i_l \left(1 - x_2 \right) + i_v \left(x_2 \right) - \bar{i} \right] \tag{20}$$

where,

$$\Psi = \left[1/\left(2\right)^{1/2} \pi \right] exp\left[-1/2 \left(v_{11}/\sigma_v' \right)^2 \right] - 1/2 \left(v_{11}/\sigma_v' \right) erfc\left(v_{11}/\left(2\right)^{1/2} \sigma_v' \right)$$

$$i_b = 0.462 \left(K \right)^{0.6} \left(Re \right)^{-0.1} \left(D_b/D \right)^{0.6} \left[1 + a \left(\rho_l - \rho_v \right) / \rho_v \right]$$

and,

$$\bar{i} \textit{ is the average fluid enthalpy}$$

$$x_2 \textit{ is the average quality in the boundary layer}$$

$$K = 2.4 \quad (\textit{experimentally determined})$$

$$D_b = \left[0.015 \left(\sigma D_h / \tau_w \right)^{1/2} \right] \left\{ 1 + 0.1 \left(g/g_c \right) \left[\left(\rho_l - \rho_v \right) / \tau_w \right] D_h \right\}^{-1/2}$$

$$v_{11} = q'' i_{fg}\left(x_2\right) / \left[\left\{i_l\left(1 - x_2\right) + i_v\left(x_2\right) - \bar{i}\right]\left(\rho_v i_{fg}\right)\right\}$$

$$\sigma_v' = \left(G/\rho_l\right) i_b$$

$$a = 0.87\left(1.36 - V_{SL}\right) + 0.135 \quad 0.5 < V_{SL} < 1.36\ m/s$$

$$a = 0.135 \qquad G < 9.7\,x\,10^6 kg/m^2\,hr,\ V_{SL} \geq 1.36\ m/s$$

$$a = 0.135\left(G/9.7\,x\,10^6\right)^{-0.3} \qquad G > 9.7\,x\,10^6 kg/m^2\,hr$$

where V_{SL} is the superficial velocity of two-phase mixture. The correlation requires iteration on q''_{CHF} since v_{11} is dependent on q''. Andreyev, et al. (Ref. 197) suggested that CHF that occurs in the transition from bubbly to dispersed-annular flow can be based on a transition void fraction and recommended an equation for this specific void fraction that agreed reasonably well with experimental data for subcooled water flow boiling. Povarnin (Ref. 198) assumed a concept of corresponding states and derived a correlation that agreed reasonably well with subcooled CHF data for various fluids including water. No further work has been performed in recent years to substantiate Povarnin's approach. Bergles (Ref. 8) identified several other proposed mechanism-based correlations for subcooled flow boiling CHF. Specifically, Chang (Ref. 199) developed a correlation for CHF based on the idea that large bubbles are broken up in a hydrodynamic limit thereby reducing the heat transfer coefficient. Thorgerson, et al., (Ref. 200) proposed a correlation which allowed a burnout condition as a result of a critical friction factor. His correlation falls somewhat in the class of expressions based on boundary layer separation. Both of these correlations have met with little success in experimental verification or general acceptance.

Probably the most widely verified mechanism-based correlation uses the superposition approach originally proposed by Rohsenow for fully developed nucleate flow boiling and extended to CHF prediction by Gambill (Ref. 201):

$$q''_{CHF} = \left[q''_{CHF}\right]_{pool\ boiling} + \left[q''_{CHF}\right]_{forced\ convection} \tag{21}$$

where,

$$\left[q''_{CHF}\right]_{pool\ boiling} = K i_{fg}\,\rho_v\left[\sigma g_c g\left(\rho_l - \rho_v\right)/\rho_v^2\right]^{1/4}\left[1 + \left(\rho_l/\rho_v\right)^{0.923}\left(c_p \Delta T_{sub}/25 i_{fg}\right)\right]$$

in which

$$K = 0.12-0.17, c_p \textit{ is evaluated at } T = T_{sat} - \left(\Delta T_{sub}/2\right)$$

$$\left[q''_{CHF}\right]_{forced\ convection} = h_{conv}\left[\left(T_w\right)_{CHF} - T_b\right]$$

and where

in which $(T_W)_{CHF}$ is evaluated with Bernath's (Ref. 202) generalized plot of wall superheat at burnout (Fig. 40). Gambill found that the correlation predicted burnout for a large number of experimental data points within a maximum deviation of 40 percent for various fluids and 17.8 percent for water. The data compared were for numerous fluids and flow configurations with a range of conditions of velocity: 0 to 174 fps, pressure: 4 to 3000 psia, subcooling: 0 to 506 °F, acceleration: 1 to 57,000 g., CHF: 0.1×10^6 to 37.4×10^6 Btu/hr ft^2. Levy (Ref. 203) presented a similar superposition correlation for CHF:

$$q''_{CHF} = q''_P + q''_C + q''_F \tag{22}$$

where,

$$q''_P = 0.131\, i_{fg}\, \rho_v \left[\sigma g_c^2\left(\rho_l - \rho_v\right)/\rho_v^2\right]^{1/4}$$

$$q''_C = 0.696\left(k_l \rho_l c_l\right)^{1/2}\left(\rho_l - \rho_v/\sigma\right)^{1/4}\left[\sigma g_c^2\left(\rho_l - \rho_v\right)/\rho_v^2\right]^{1/8}\Delta T_{sub}$$

and,

$$q''_F = h_l\left(\Delta T_{sat}\right) + h_l \Delta T_{sub}$$

where h_l is determined from a pure forced convection Nusselt number,

$$h_l = Nu\, k_l / D$$

and $(\Delta T_{sat})$is determined by trial and error from a fully developed nucleate boiling relation such as Jens and Lottes (Eqn. 11) at q''_{CHF}.

The list of mechanism-based CHF correlations are by no means limited to those presented here. Those for which the equations are presented appear to be generally applicable to highly subcooled, water flow boiling which will be encountered in the cooling of high enthalpy arc facility components. A statement made by Bergles in Ref. 8, which is applicable here, is that there are enough "adjustable" constants in any predictive equation to permit an acceptable correlation of data.

4.5.2 Dimensional Analysis/Similitude-based CHF Correlations

Though dimensional analyses have been performed by Barnett (Ref. 204), Zenkevich (Ref. 205), and Kampfenkel (Ref. 206), Glushchenko (Ref. 207) used partial modeling to obtain a correlation for highly subcooled (greater than 25 °C) water flow in tubes and annuli based on the following dimensionless groups:

$$K_1 = q''_{CHF}/i_{fg}\,\rho_v V_{avg}$$

$$K_2 = c_p \Delta T_{sub}\,\rho_l / i_{fg}\,\rho_v$$

$$K_3 = k_l / V_{avg} D_e c_p \rho_l$$

$$K_4 = i_{fg}/c_p \Delta T_{sat}$$

Glushchenko proposed the correlation,

$$K_1 = 18.25 K_2^{0.35}\, K_3^{0.5}\, K_4^{1.2} \tag{23}$$

which agreed reasonably well with experimental data (80 percent of nearly 200 selected data points correlated within 25 percent). The data covered the ranges: mass velocity from 500 to 40,000 kg/m^2 sec (100 to 8200 lbm/ft^2 sec), pressure from 4.9 x 10^5 to 197 x 10^5 N/m^2 (72 to 2900 psia), subcooling from 25 to 250 °C (77 to 482 °F), diameter from 2 to 12 mm (0.08 to 0.5 in.), and L/D from 10-20 to 60-120. Ornatskii, et al. (Ref. 208), later extended the approach to low subcooling and moderate quality water flow.

Probably the most successful approach to similitude analysis of the CHF phenomena was performed by Ahmad (Ref. 68). The primary impetus for such analyses was the need for fluid-to-fluid modeling capability. Refrigerants such as Freon have been used in previous experiments (Refs. 64-67) to simulate water cooling at high heat flux conditions since the CHF for Freon can be achieved at much lower heat flux. Scaling methods were necessary to provide a means of relating the Freon test results to water for high heat flux/water cooling predictions. Ahmad used classical dimensional analysis to identify 12 independent groups, of which 6 were eliminated through inductive arguments. Of the remaining six terms, 3 terms (subcooling number $\Delta i/i_{fg}$, liquid/vapor density ratio ρ_1/ρ_v , and L/D) can be independently satisfied in a controlled experiment (controlled inlet temperature, pressure, and geometry). The remaining 3 terms (Reynolds number, Weber-Reynolds number, and liquid/vapor viscosity ratio) are distorted through the liquid and vapor viscosities when the above conditions are fixed. Ahmad solved this problem of multiple distortion by expressing the remaining 3 terms as a CHF modeling parameter Ψ_{CHF}, and using a compensated distortion model to relate the terms through two empirical exponents. The final form of the modeling parameter is written as,

$$\Psi_{CHF} = \left[\left(GD/\mu_l\right)\left(\mu_l^2/\sigma D\rho_l\right)^{2/3}\left(\mu_v/\mu_l\right)^{1/5}\right] \tag{24}$$

From the Buckingham Pi theorem the dependent parameter, boiling number (q''_{CHF}/Gi_{fg}), is written as follows:

$$q''_{CHF}/Gi_{fg} = f\left(\Psi_{CHF}, \Delta i_i/i_{fg}, \rho_l/\rho_v, L/D\right) \tag{25}$$

In addition to the independent parameters identified above, Ahmad included the ratio of the heated equivalent diameter (4 x flow area/heated perimeter) to the hydraulic diameter (4 x flow area/wetted perimeter), D_{he}/D, for more complex geometries. Ahmad demonstrated reasonable scaling of subcooled Freon CHF data from Coffield, et al. (Ref. 209) to water CHF using the above approach. Analysis of complex geometries by this method is limited by the modeling parameter in which the empirical exponents are highly sensitive to geometric parameters. Use of the correlation for situations other than extrapolation of refrigerant CHF data to water CHF has been very limited (see, for example, Katto - Ref. 210).

4.5.3 Empirical-based CHF Correlations

Probably the most common approach to obtaining a CHF correlation is the curvefitting of experimental test data, although some confusion exists as to whether or not a correlation is considered empirical when some or all of the parameters used in the correlation are obtained from dimensional analysis or limited boiling mechanism analyses. Because most correlations rely on some form of empiricism, this is probably a moot point and some overlap of the mechanism-based, dimensional analysis/similitude-based, and empirical-based approaches exists. A word of caution about the use of empirical-based correlations: the expressions typically have been derived for a specific range of experimental conditions, and extrapolation of a correlation outside the specific range can lead to very large errors.

The earliest empirical CHF correlations appeared in the late 1940's into the early 1950's, most notably those of McAdams, et al. (Ref. 24), Jens and Lottes (Ref. 85), Gunther (Ref. 27), Buchberg, et al. (Ref. 97), and McGill and Sibbitt (Ref. 211). The correlations proposed by McAdams' team and Gunther are limited to low pressure and only McGill and Sibbitt evaluated subcoolings greater than 250 °F. All of the correlations are limited to low coolant velocity (less than 40 fps or a mass velocity less than 2500 lbm/ft sec). Only the Jens and Lottes correlation included a pressure effect and none included geometry effects. The test configurations were typically tubes except those used by McAdams' team (annulus) and Gunther (metal strip in a rectangular channel).

Weatherhead (Ref. 42) later modified the Jens and Lottes correlation to include the effects of geometry and latent heat of vaporization. Weatherhead subdivided subcooled flow boiling into four classifications and accounted for variations in boiling characteristics at low and high pressure, low and high mass velocity, small and large geometries, and low and high subcooling. The correlation at high subcooling, however, could only be validated at low L/D, and Tong (Ref. 10) later found that Weatherhead's correlation did not improve the accuracy of prediction when other data were evaluated.

A number of empirical correlations appeared in the late 1950's into the mid 1960's, many of which are summarized in Ref. 10 (see also Refs. 212 and 213). Most notable are correlations derived by Bernath (Ref. 202), Janssen, et al. (Refs. 214 and 215), Labuntsov (Ref. 216), Van Huff and Rousar (Ref. 29), and the United Kingdom Atomic Energy Establishment at Winfrith, AEEW (Refs. 81, 88, 217 and 218). Bernath developed an empirical correlation based on the premise that at CHF the two-phase flow near the heated surface is highly turbulent and, therefore, well mixed. Assuming the convective heat transfer through this mixture, Bernath empirically formulated expressions for the wall superheat and the heat transfer coefficient at the critical point (Ref. 219). He later (Ref. 202) refined the correlation for an extended range of geometric variables, and for water is given as,

$$q''_{CHF} = h_{CHF}\left(T_{w,CHF} - T_b\right)) \qquad (26)$$

where,

$$T_{w,CHF} = 57\, ln\, p - 54\left|p/\left(p + 15\right)\right| - V/44$$

$$h_{CHF} = 19{,}602\left|D_e/\left(D_e + D_i\right)\right| + \left(slope\right)VV$$

$$\begin{aligned} slope \; = \;& 96.4/D_e^{0.6} && for\, D_e < 0.1\, ft \\ & 162 + (18/D_e) && for\, D_e \geq 0.1\, ft \end{aligned}$$

where h_{CHF} is in Btu/ft^2 hr °C, $T_{w,CHF}$ and T_b are in °C, and D_i is the heated perimeter (inner diameter) in ft divided by π. Bernath found that the correlation predicted the CHF for nearly 250 forced water flow experiment data points within 16 percent. The range of parameters for the correlation are: pressure from 23 to 3000 psia, velocity from 4 to 54 fps, subcooling from 0 to 615 °F, and hydraulic diameter from 0.143 to 13.0 in. Multiple configurations including tubes, annuli, ducts, and ribbons were used in the experiments.

Janssen and others (Refs. 214 and 215) developed rather cumbersome design correlations (presented in Ref. 10) for CHF in subcooled and saturated water flow in tubes and annuli (inner wall heated). Their correlations are valid over the parameter ranges: pressure from 600 to 1450 psia, mass velocity from 0.2 to 6.2 lbm/ft^2 hr, and hydraulic diameters from 0.25 to 1.25 in. Labuntsov (Ref. 216) assumed (in the first approximation) that the CHF is dependent on pressure, velocity, and subcooling, and developed a CHF correlation for water flow at pressures up to 200 atm. Van Huff and Rousar (Ref. 29) compiled CHF data on 23 different fluids, including water, during the study of heat flux limits of storable propellants. They correlated the higher $V\Delta T_{sub}$ water data as shown in Fig. 7 with the expression,

$$q''_{CHF} = 5.1 + 0.000860\, V\Delta T_{sub} \tag{27}$$

where q''_{CHF} is in Btu/in.^{2} sec . Considerable scatter was noted at low $V\Delta T_{sub}$ (less than 10,000 ft °F/sec) and a significant pressure effect was observed for data obtained in swirl flow. The correlation is valid over the range of parameters: pressure from 10 to 2000 psia, velocity from 7.5 to 205 fps, and bulk fluid temperature from 76 to 470 °F.

Lee and Obertelli (Ref. 81) at AEEW in the UK satisfactorily correlated early AEEW subcooled flow boiling CHF data obtained at 1000 psia using a modified version of an widely used correlation derived by DeBortoli, et al. (Ref. 220) for higher pressure. Lee and Obertelli also found that a more general correlation (based on a local conditions hypothesis) under consideration by AEEW colleagues at about the same period of time (see Barnett - Ref. 221) provided excellent accuracy in the prediction of over 600 CHF data points obtained at AEEW over a range of pressure from 560 to 1600 psia. Macbeth (Ref. 217) and, later, Thompson and Macbeth (Ref. 218) correlated nearly 4400 CHF data points (with a rms error of 7.5 percent) obtained for water flow boiling in round tubes with expressions based on Barnett's local condition hypothesis. The hypothesis assumes that the CHF is only a function of the mass quality at the point of overheating. Macbeth showed that a nearly linear relationship exists at low mass velocity, therefore, he divided the boiling map into a low velocity regime and a high velocity regime. Figure 41 presents the boundary limits of the two regimes as identified by Macbeth. The CHF correlations for the two regimes as given by Thompson and Macbeth are,

High Velocity Regime

$$q''_{CHF} = \left\{\left[A' + 0.25D\left(G\,x\,10^{-6}\right)\Delta i_i\right] / \left(C' + L\right)\right\} x\,10^6 \tag{28}$$

where A' and C' are given in Fig. 42,

Low Velocity Regime

$$q''_{CHF} = \left\{\left[\left(G\,x\,10^{-6}\right)\left(i_{fg} + \Delta i_i\right)\right] / \left[158\,D^{0.1}\left(G\,x\,10^{-6}\right)^{0.49} + 4L/D\right]\right\} x\,10^6 \tag{29}$$

The terms L and D are expressed in inches. Barnett (Ref. 88) later modified the basic form of these equations to obtain a correlation more applicable to annulus data,

$$q''_{CHF} = \left|\left(A + B\Delta i_i\right) / \left(C + L\right)\right| x\, 10^6 \tag{30}$$

where,

$$A = 67.45\, D_{HE}^{\;0.68}\left(G\, x\, 10^{-6}\right)^{0.192}\left\{1 - .0744\, exp\left|-6.512\, D_{HY}\left(G\, x\, 10^{-6}\right)\right|\right\}$$

$$B = 0.2587\, D_{HE}^{\;1.261}\left(G\, x\, 10^{-6}\right)^{0.817}$$

$$C = 185.0\, D_{HY}^{\;1.415}\left(G\, x\, 10^{-6}\right)^{0.212}$$

$$D_{HE} = \left(D_O^2 - D_I^2\right)/D_I$$

$$D_{HY} = D_O - D_I$$

With the correlation above, Barnett correlated 724 annulus CHF data points with a rms error of 5.9 percent. Limits of the correlation are: D_I from 0.375 to 3.798 in., D_O from 0.551 to 4.006 in., L from 24 to 108 in., G x 10^{-6} from 0.14 to 6.2 lbm/hr ft^2, and Δi_i from 0 to 412 Btu/lbm.

A number of empirical CHF correlations developed for subcooled flow boiling have appeared since the 1960's. Tolubinsky, et al. (Ref. 137), at the Institute of Engineering Thermophysics, USSR, proposed a design equation for determining CHF in annular channels with the inner wall heated, noting that because the CHF is affected by the heated surface shape and dimensions, the CHF data obtained from tube flow cannot be applied to annular flow. Their expression correlated more than 90 percent of approximately 400 experimental data points to within ± 25 percent. The correlation, however, requires knowledge of the CHF during pool boiling conditions for the annular configuration of interest, which may hinder its usefulness. More recent work performed in the USSR regarding CHF in flow boiling correlations include the studies by Levitan and others (Refs. 222 and 223) and the Scientific Council of the Academy of Sciences of the USSR (Ref. 224). Levitan and his team proposed the following correlation for subcooled and low-quality water flow in tubes,

$$q''_{CHF} = \left|10.3 - 7.8\left(p/98\right) + 1.6\left(p/98\right)^2\right| e^{-1.5x}\left(G/1000\right)^{1.2\{0.25\{(p/98) - 1\} - x\}} \tag{31}$$

where p is in bars, G is in kg/m^2 sec, and x is the relative quality at CHF. The correlation is good for water flow in circular tube with a diameter within the range of 4 to 16 mm (0.16 to 0.63 in.). Levitan and his team, along with several other researchers, later assisted the Scientific Council of the Academy of Sciences of the USSR (Ref. 224) in compiling tabular data for predicting CHF in uniformly heated, 8-mm (0.31-in.) diameter tubes. An approximation formula is provided for calculating CHF for tubes whose diameter is different than 8 mm:

$$q''_{CHF} = q''_{CHF,8}\left(8/D\right)^{0.5} \tag{32}$$

where $q''_{CHF,8}$ is the value of q''_{CHF} in the 8-mm diameter tube and D is in mm. The tabular data are quoted to have a rms error of 10 percent. The tabular correlation is limited to the following range of conditions: pressure from 29.5 to 196 bar (435 to 2880 psia), mass velocity from 750 to 5000 kg/m^2 sec (154 to 1024 lbm/ft^2 sec), subcooling from 0 to 75 °C (32 to 167°F), diameter from 4 to 16 mm (0.16 to 0.63 in.), and length to diameter ratio of 20 or greater. Doroschuk, et al.(Ref. 222) modified Eqn. (31) to include a variable critical pressure such that the correlation could be used for other fluids.

Bowring (Ref. 225) built upon the work of Thompson and Macbeth (Ref. 218) and developed a CHF correlation using four basic variables as functions of pressure. He showed that the correlation predicted approximately 3800 data points with a rms error 6.96 percent. At about the same time, Becker, et al. (Ref. 226) proposed a CHF correlation for round tubes (their data were for primarily 10-mm diameter tubes). Their correlation is given as,

$$q''_{CHF} = \left[G\left(450 + \Delta i_i\right)/\left(40L/D + 156G^{0.45}\right)\right]\left[1.02 - \left(p_r - 0.54\right)^2\right] \tag{33}$$

where D and L are expressed in meters, Δi_i in kJ/kg, G in kg/m^2 sec, and p_r in bars. The expression correlated over 500 data points to a rms error of 5.7 percent and is good for the following range of conditions: pressure from 120 to 200 bar (1765 to 2940 psia), mass velocity from G(p), as given in Fig. 42, to 7000 kg/m^2 sec (up to 1435 lbm/ft^2 sec), inlet subcooling from 8 to 272 °C (46 to 522 °F), heated length from 2000 to 5000 mm (79 to 197 in.), and steam quality from -0.3 to 0.6, where negative quality represents a subcooled liquid.

Knoebel, et al. (Ref. 78) developed a CHF correlation for subcooled water flow based on previous work performed at the Savannah River Laboratory:

$$q''_{CHF} = K\left(1 + 0.0515V\right)\left(1 + 0.069\Delta T_{sub}\right) \tag{34}$$

where K equals 153,600 for stainless steel heaters. The correlation is limited to low-pressure water flow (less than 45 psia) and higher subcooling (greater than 45 °F).

Green and Lawther (Ref. 227) developed a CHF correlation for high pressure water as well as Freon,

$$q''_{CHF} = Re_v^n \left(\rho_l / \rho_v\right)^m \cdot Pr_v^w \sigma_N^p f\left(L_{sat}/D\right)\left(1 + \delta\right)\left(1 - \delta_l\right) \tag{35}$$

where,

$$n = 1 - exp\left[-0.0067\left(L_{sat}/D\right)\right]$$

$$m = 0.1 + exp\left[-0.007\left(L_{sat}/D\right)\right]$$

$$p = -0.5\left\{0.15 + exp\left[-0.007\left(L_{sat}/D\right)\right]\right\}$$

$$w = -\left\{0.21 + 0.55\, exp\left[-0.007\left(L_{sat}/D\right)\right]\right\}$$

$$f\left(L_{sat}/D\right) = 9x10^{-5} exp\left\{-0.00055\left(L_{sat}/D\right) + 3.83\, exp\left[-0.00396\left(L_{sat}/D\right)\right]\right\}$$

$$\delta = exp\left\{-\left[0.14x10^8 \sigma_N + 0.02\left(L/D\right)Pr_v\right]\right\}$$

$$\sigma_N = \sigma / \left(D \Delta i_v \rho_v\right)$$

$$\delta_l = 0.75\, exp\left\{-B\left(L/D\right)\left(V_m \sigma / \rho_l \mu_l \Delta i_v\right)\right\}$$

$$B = 130.5\, exp\left\{5.0\, exp\left(-0.02\, L/D\right)\right\}$$

The expression was quoted to have correlated over 4250 data points with a rms error of less than 10 percent, although the correlation proposed by Thompson and Macbeth (Ref. 218) predicted the higher pressure data better. The correlation is limited to reduced pressure less than 0.7, mass fluxes greater than 200 kg/m^2 sec (41 lbm/ft^2 sec), and exit quality greater than 0.1.

Finally, Shah presented graphical correlations for CHF in subcooled and saturated flows in tubes (Ref. 228) and annuli (Ref. 229). The tube correlation in functional form is,

$$Bo = f_1\left(x_{in}, L/D\right) \qquad for\, Y < 10^4$$

$$Bo = f_2\left(x_{cr}, Y, p_r\right) \qquad for\, Y > 10^5 \tag{36}$$

where,

$$Y = \left(GDc_p / k_l\right)\left(G^2/\rho_l^2 gD\right)^{0.4}\left(\mu_l/\mu_v\right)^{0.6}$$

The graphical expressions correlated 90 percent of 1271 tube data points within ±30 percent. The data include various fluids including water at reduced pressure from 0.0012 to 0.94, mass flux from 6 to 24,300 kg/m2 sec (1.2 to 4980 lbm/ft2 sec), and inlet quality from -3.0 to positive values. The annulus correlation in functional form is,

$$Bo = f_3\left(x_{in}, L/D_{hp}\right) \tag{37}$$

where,

$$D_{hp} = 4\, x\, Flow\, area\, /\, Heated\, perimeter$$

The graphical expression correlated 88 percent of 825 annulus data points and one non-circular geometry data point within ±30 percent. The data include various fluids including water at reduced pressure from 0.017 to 0.9, mass flux from 100 to 15,780 kg/m2 sec (20 to 3230 lbm/ft2 sec), heated perimeter diameter from 5.3 to 96.3 mm (0.2 to 3.8 in.), gap width from 0.5 to 11.1 mm (0.02 to 0.44 in.), L/D_{hp} from 3.7 to 335, and inlet quality from -3.1 to 0.0.

4.5.4 Miscellaneous CHF Correlations

The CHF correlations discussed thus far have been for internal flows with steady-state, uniform heating. Certainly numerous correlations for external flows also exist. The reader is referred to Lienhard (Ref. 33) for an excellent discussion of CHF correlations for external flow. Very little work has been performed in deriving an expression for CHF that accounts for transient effects. Kataoka, et al. (Ref. 123) proposed a sixth dimensionless group (to account for transient effects) in addition to the five groups proposed by Katto (Refs. 184 and 186). They developed the following CHF correlation for power transients:

$$q''_{CHF,pwr\ tr} = 0.3740\, G\, i_{fg} \left(\rho_v / \rho_l\right)^{0.66} \left(\sigma\, \delta_l / G^2 l_o\right)^{0.40}$$

$$x \left\{ 1 + \left[0.03648 \left(l_o / D_{he} \right)^{-0.20} \left(\rho_v / \rho_l \right)^{-0.79} + \varepsilon \right] \Delta i_i / i_{fg} \right\}$$

$$+ 0.2038 \left(\rho_v / \rho_l \right)^{0.52} \left(\sigma\, \rho_l / G^2 l_o \right)^{0.19} \left(\tau G / \rho_l l_o \right)^{-0.63} \tag{38}$$

where,

$$\varepsilon = 0 \qquad for\, \Delta T_{sub} = 0 - 40\ °K$$

$$\varepsilon = 0.00808 \left(\rho_v / \rho_l \right)^{-1.09} \left[\left(\Delta i_i - \Delta i_{i40} \right) / \Delta i_i \right] \qquad for\, \Delta T_{sub} = 40 - 70\ °K$$

$$l_o = \left\{ \sigma / \left[g \left(\rho_i - \rho_v \right) \right] \right\}^{1/2}$$

$$D_{he} = \left(D_{ts}^2 - D_{htr}^2 \right) / D_{htr}$$

$$\tau = exponential\ period\ of\ power\ increase$$

and Δi_{i40} is the enthalpy difference between the local and 40 °K inlet subcooling. Kataoka's team correlated a large number of data points with the above expression to within ±20 percent. It should be noted that the data were obtained at low velocity (4.43 to 13.26 fps) for water flowing lengthwise along a platinum wire. Other experimental conditions were: pressure from 0.143 to 1.503 MPa (21 to 221 psia), subcooling from 0 to 70 °K (0 to 126 °F), power transient periods from 5 to 500 msec, diameter from 0.8 to 15 mm (0.03 to 0.6 in.), and wire length from 3.93 to 10.04 cm (1.5 to 4 in.). Celata, et al. (Ref. 126) proposed the following critical mass flow correlation for flow transient effects,

$$\left(\Phi_{m,CHF} \right)_{tr} = \left(\Phi_{m,CHF} \right)_{ss} \left\{ 1 - exp \left[-0.87 \left(p_r \right)^{-0.57} \left(\tau \right)^{0.54} \left(q''/q''_{CFH,ss} \right)^{2.8/\tau} \right] \right\} \tag{39}$$

where q" is in W/m^2, τ is the ratio between the half-flow decay time, $t_{\dot{m}/2}$, and the time transit parameter, t_{tr}. Celata's team correlated 82 percent of their data (approximately 400 data points) with the expression to within ±20 percent. Their data were obtained with Refrigerant-12 flow in 7.5-mm (0.3-in.) diam, 2300-mm (91-in.) long tubes at the following conditions: pressure from 1.2 to 2.75 MPa (176 to 404 psia), heat flux from 32000 to 85000 W/m^2(10200 to 27000 Btu/ft^2 hr), inlet subcooling of 23 °C (73 °F), and half-flow decay time from 0.4 to 10 sec.

An excellent discussion of non-uniform heat flux distribution effects on CHF is included in Chapter 9 of Ref. 7. Three approaches to correlating non-uniform heating data are presented: the 'local conditions' hypothesis, the 'overall power' hypothesis, and the 'F-factor' method. No general analytical method exists for the prediction of CHF in configurations with non-uniform heat flux distribution. Collier (Ref. 7) recommends the 'F-factor' method as the best purely empirical approach for prediction of non-uniform heat flux CHF. The method involves an energy balance on the superheated boundary layer in the bubbly flow region to obtain a correction factor, F, which is defined as the ratio of the CHF at any given local enthalpy for the uniform heat flux profile case to the CHF at the same given local enthalpy for the particular non-uniform heat flux case. The procedure for predicting CHF using the 'F-factor' method is given on pp. 268-269 of Ref. 7.

Practical limits to CHF can be determined from a simple heat transfer analysis of the critical heat flux condition (Ref. 7). The critical heat flux condition cannot exist if the heater surface temperature lies below the saturation temperature. The minimum possible CHF is then given by

$$q''_{CHF,min} = \left(\Delta T_{sub}\right)_i / \left[4L/\left(G c_l D\right) + 1/h\right] \tag{40}$$

In addition, the critical heat flux condition must occur at or before all of the fluid in the channel is evaporated (quality equals unity) which leads to the maximum possible CHF,

$$q''_{CHF,max} = \left(G D i_{fg} / 4L\right)\left[1 + c_l\left(\Delta T_{sub}\right)_i / i_{fg}\right] \tag{41}$$

Gambill and Lienhard (Ref. 230) presented an interesting semiempirical approach to determining a practical upper limit to CHF. From kinetic theory they derived an expression for $q_{max,max}$, the highest heat flux that can conceivably be achieved in a phase-transition process,

$$q''_{max,max} = \rho_g i_{fg} \left(RT/2\pi\right)^{1/2} \tag{42}$$

A practical upper limit to heat transfer by phase change was shown to be $0.1q_{max,max}$; however, the method is limited to pressures less than about one tenth of the critical pressure. As a point of interest, the highest CHF achieved for any configuration and test condition was nearly 3.4 x 10^8 W/m^2 (30,000 Btu/ft^2 sec) measured by Ornatskii and Vinyarskii (Ref. 231) with a nonuniformly-heated (circumferential), small-bore tube.

4.6 COMPARISON OF FLOW BOILING AND CHF CORRELATIONS

Guglielmini, et al. (Ref. 170) presented an interesting comparison of flow boiling correlations including boiling incipience and partial nucleate boiling. Most of the boiling incipience correlations, when compared to representative data, had an accuracy within 30 percent. A quantitative comparison of the partial nucleate boiling expressions could not be performed because of lack of data. The correlations for fully developed nucleate boiling generally predicted representative data to within ±30 percent, although in some cases the data scatter caused error in the prediction to be as large as 70 percent.

Gambill (Ref. 232) compared ten CHF correlations for water flow in a tube (diameter of 0.1 in. and length of 20 in.) at a pressure of 600 psia and subcooling of 100 °F over a range of velocity from 10 to 80 fps. Prediction of CHF at the higher velocity using the correlations varied by nearly a factor of four and by a factor of two at the lower velocity. Factors of two or greater between correlation predictions at higher mass velocity have also been noted by Zeigarnik, et al. (Ref. 93) and Boyd (Ref. 17), and the disagreement tends to worsen as mass velocity increases. It should be reiterated that empirical CHF correlations typically have been derived for a specific range of experimental conditions, and extrapolation of a correlation outside the specific range can lead to very large errors. In addition, a given analytical CHF correlation typically has enough "adjustable" constants in the predictive equations to permit an acceptable correlation of a specific data set; however, general application of the correlation to various configurations or test conditions may result in significant errors.

5.0 APPLICABLE EXPERIMENTAL APPROACHES AND MEASUREMENT TECHNIQUES FOR BOILING HEAT TRANSFER

A representative list of flow boiling experiments that have been performed within the last 50 years is presented in the Appendix. A few comments concerning the experiments listed in the Appendix are in order. The heating method of choice is electrical resistance heating (Joule heating). A fewer number of experiments made use of cartridge heaters, and very few incorporated some other means of heating. The test sections were primarily tubes, although a significant number of experiments had annular test sections. Of the more than 130 listed experiments where water was used, in only 23 was the maximum heat flux greater than 1000 Btu/ft^2 sec, and in only 6 was it greater than 5000 Btu/ft^2 sec, with most of these performed at a pressure less than 500 psia, subcooling less than 300 °F, or velocity less than 100 fps. Clearly, data obtained at high heat flux, pressure, subcooling, and velocity are severely limited.

A number of measurement techniques used in these and other experiments are worth reviewing. One of the most important measurements in a boiling experiment is surface temperature. Placement of a discrete, intrusive temperature measuring device such as a thermocouple on the surface at the fluid interface may disrupt the flow field, causing large errors in the temperature measurement. For this reason, backside wall temperature is typically measured and used to estimate surface/fluid interface temperature through the use of an analytical temperature distribution. On an electrically heated surface, the temperature measurement with a thermocouple (which provides temperature from millivolt changes) is further complicated by voltage drop in the test specimen due to current flow. Hughes (Ref. 21) incorporated a single wire junction at the surface and the two wire thermocouple junction formed 1/32 inch from the surface, thereby preventing any extraneous voltages due to voltage drop between the thermocouple leads. Numerous researchers have used thin sheets of mica or other electrical insulating material to isolate the thermocouple from the current-carrying surface. One of the most attractive approaches as suggested by Dutton and Lee (Ref. 233) involves the use of a three wire thermocouple. The thermocouple circuit is "balanced" by applying and subsequently reversing the current and nullifying the voltage drop with a voltage-dividing potentiometer, the net effect being the same as welding all three wires to the location of the middle wire.

Burnout detectors have been used by a number of investigators (e.g., Refs. 21, 234, and 235) to protect the test sections from destruction as the CHF is approached. However, as pointed out previously, Hughes (Ref. 21) noted that the speed and intensity of the transition to film boiling (i.e., burnout) was observed to increase with increasing velocity and subcooling, and in some cases burnout has occurred instantaneously (Ref. 145). Therefore, the effectiveness of a burnout detector to protect the test section in highly subcooled, high velocity, high pressure flow is questionable.

Gunther (Ref. 27) was one of the earliest to perform a detailed photographic study of subcooled flow boiling at higher heat flux, although pressure was limited to below 164 psia and velocity limited to below 40 fps. His photographic system was capable of 20,000 frames/sec and gave a resolution of bubbles as small as 0.004-in. diameter. Fiori and Bergles (Ref. 20) used several photographic techniques, including high speed photographs, movies and video. They successfully identified cyclic variations in surface temperature from variations in the flow structure by simultaneously recording surface temperature and high speed photographs of a specific surface location. Ram, et al. (Ref. 236), though studying bubble generation in pool boiling of water, analyzed intensity fluctuations registered by a photomultiplier tube to determine bubble size and growth cycle. Lineberger (Ref. 237) used a dual frequency sound field to determine bubble size in pool boiling of water. Brown (Ref. 39) used reflected light from bubbles detected by a light dependent resistor to identify the presence of first vapor when boiling begins.

Several approaches have been used to determine the vapor thickness or void fraction near the heated surface in local boiling. Costello (Ref. 238) measured the number of beta particles emitted from a vial containing radioactive strontium 90 to determine the vapor thickness in a low velocity, water flow boiling apparatus with an annular test section. Rogers, et al. (Ref. 239) used a traversing gamma-ray densitometer (cobalt-57 source) to measure void fractions in water flow boiling in an annulus. Similar densitometers have been used by Buchberg, et al. (Ref. 97) and Edelman, et al. (Ref. 240) to measure the void fraction of subcooled water flow in tubes. Jiji and Clark (Ref. 241) used a specially developed traversing thermocouple probe to measure bubble boundary layer thickness and temperature profiles in low velocity, subcooled water flow. Later, Stefanovic, et al. (Ref. 242) used a similar probe to measure temperature fluctuations within a superheated boundary layer adjacent to a heated annular wall in forced water flow.

The measurement of acoustic noise due to boiling has permitted various researchers to study hydrodynamic instabilities in heated channels and to diagnose the different regimes of boiling. Pressure transducers have typically been incorporated in an experimental apparatus where flow oscillations are to be studied (e.g., Refs. 243 and 244), although, Romberg and Harris (Ref. 245) used piezoelectric accelerometers to measure vibrations associated with parallel channel, density wave, and acoustic oscillations in a flow boiling loop. Several investigations have been performed to show that information on the boiling mode can be ascertained from boiling noise (Refs. 246-250). Hydrophones (i.e., transducers) with piezoceramic sensing elements have typically been used to measure the acoustic emissions from boiling.

A number of important considerations can be learned from the success or failure of previous experiments and associated analyses. Westwater (Ref. 11) pointed out that one basic method of proving the correctness of instrumentation in these types of experiments which is widely overlooked is a simple energy balance of the system. Accuracy of the all important wall temperature can be severely affected by overlooked items such as conduction effects or temperature dependent material properties. Flow loop layout can affect the hydrodynamic stability of the system which can seriously degrade the heat transfer and CHF measurements. Gas content and cooling fluid impurities have been shown to have an effect on boiling heat transfer and CHF, yet in many experiments these have been ignored or have not been adequately quantified. Finally, the geometric scale of a heater surface has been shown to be very important. Specifically, Bakhru and Lienhard (Ref. 251) demonstrated that the hydrodynamic processes that give rise to burnout cease to occur when the Laplace number[12], R', is reduced below a value on the order of 0.1 (see also Refs. 33 and 252). Generalized correlations based on data acquired from a configuration with a Laplace number below this value may have very large errors when applied to scaled-up or different configurations.

6.0 CONCLUSIONS

An extensive review of backside water cooling processes that are applicable to high enthalpy facility components has been conducted. The processes can be

[12] The Laplace number is defined as $R' = r[g(\rho_l - \rho_v)/\sigma]^{1/2}$, where r is a characteristic radius dimension of the heater (see Ref. 251).

identified on the basic boiling curve which, for the configurations of interest, includes the pure convection regime, the nucleate boiling regime, and the boiling crisis point where the critical heat flux (CHF) occurs. More than 20 parameters have been shown to affect various portions of the boiling curve, and optimum cooling (and greater CHF) occurs at high subcooling and high mass velocity. The primary parameters which affect steady-state boiling heat transfer are pressure, mass velocity, subcooling, heater diameter, and heater length; however, other parameters have been experimentally shown to have significant effects. Several heat transfer enhancement techniques have also been reported.

Because of the complexity, the unsteady nature, and the small scale of the flow boiling processes, no general theory has been developed. Reasonable confidence in correlating heat transfer data in the pure convection and nucleate boiling regimes is shown. The more recent pure forced convection correlations of Petukhov (Eqn. 5) for turbulent flow in a tube and Kays and Leung (Ref. 151) for turbulent flow in an annulus are probably the most useful for the nonboiling regime. Bergles and Rohsenow's partial nucleate boiling expression (Eqn. 8) best correlates the transition from pure convection to fully developed nucleate boiling, and the widely accepted Rohsenow relation (Eqn. 16 and 17) ranks among the best correlations for fully developed nucleate boiling in forced flow. Considerable disagreement between the large number of analytical and empirical CHF correlations currently exists. Moreover, because the CHF correlations are sensitive to the range of conditions for the data used, no single correlation can be recommended. Very few data exist that were obtained at high heat flux, high subcooling, high mass velocity, and high pressure, which are optimum for cooling high enthalpy facility components. Finally, numerous shortcomings of previous experiments have been identified, and care must be taken in any experimental program where boiling heat transfer data are acquired for future analysis and correlation.

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from Ref. 3

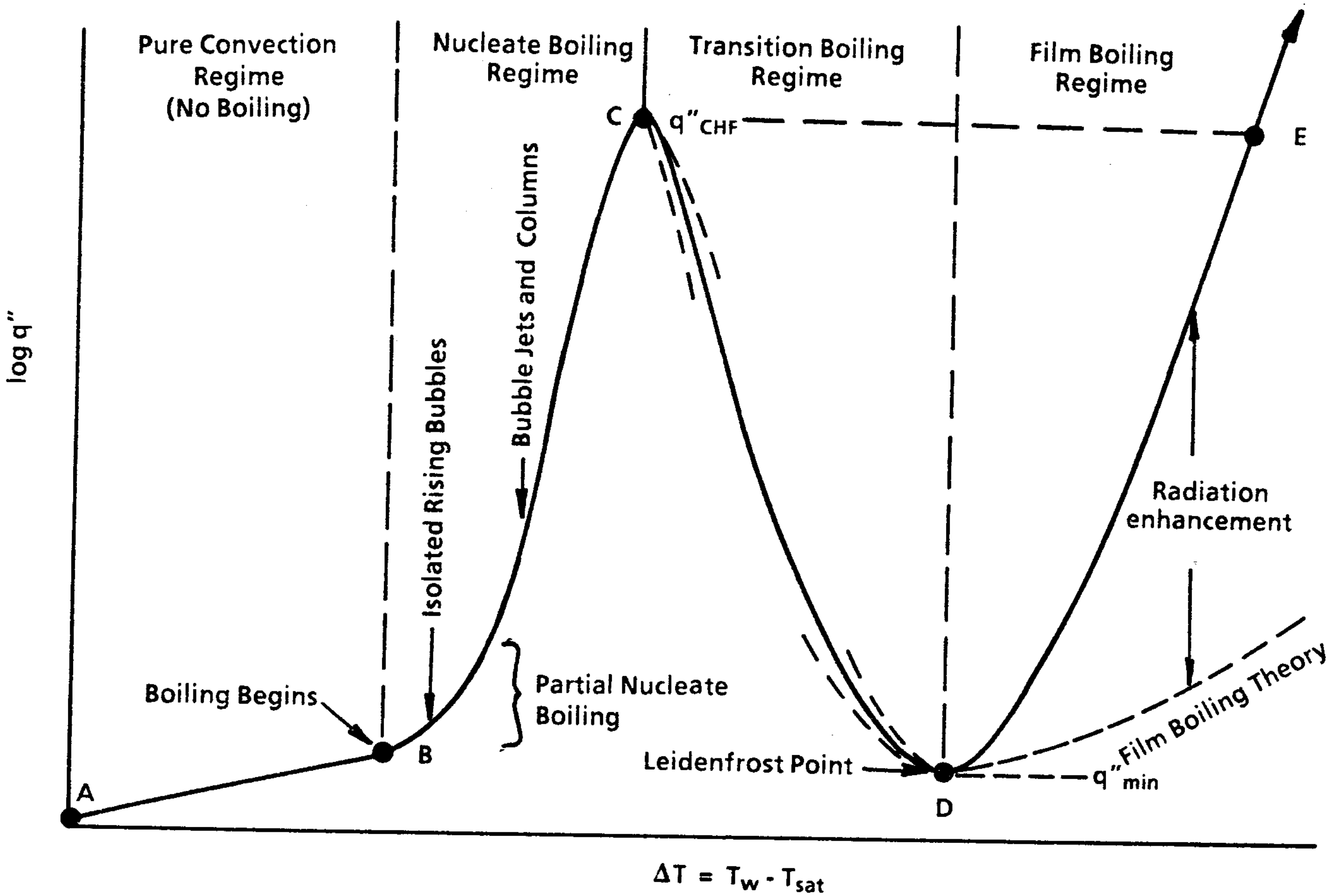


Figure 1. The Basic Boiling Curve

from Ref. 22

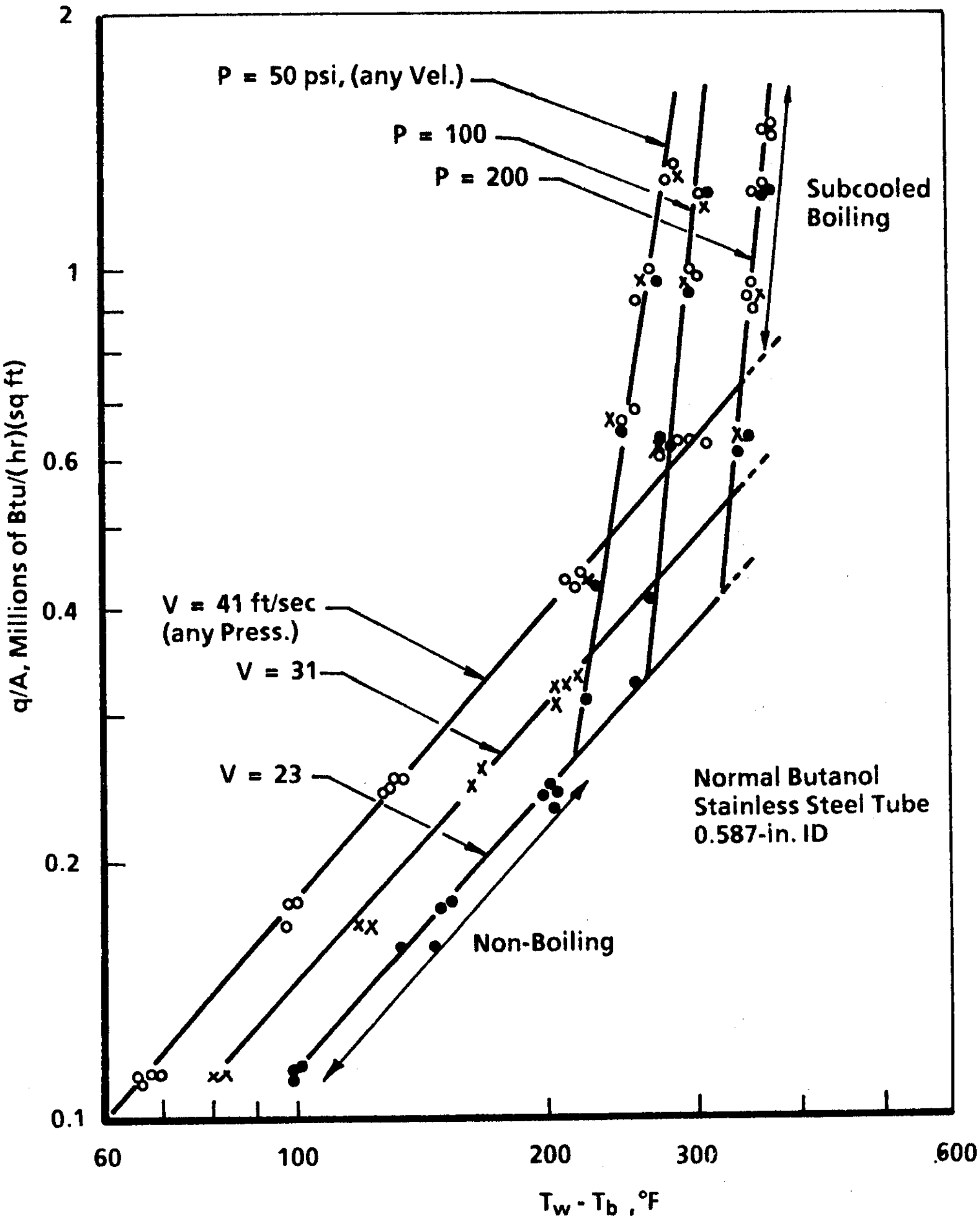


Figure 2. Effect of Pressure and Velocity on Subcooled Boiling

from Ref. 24

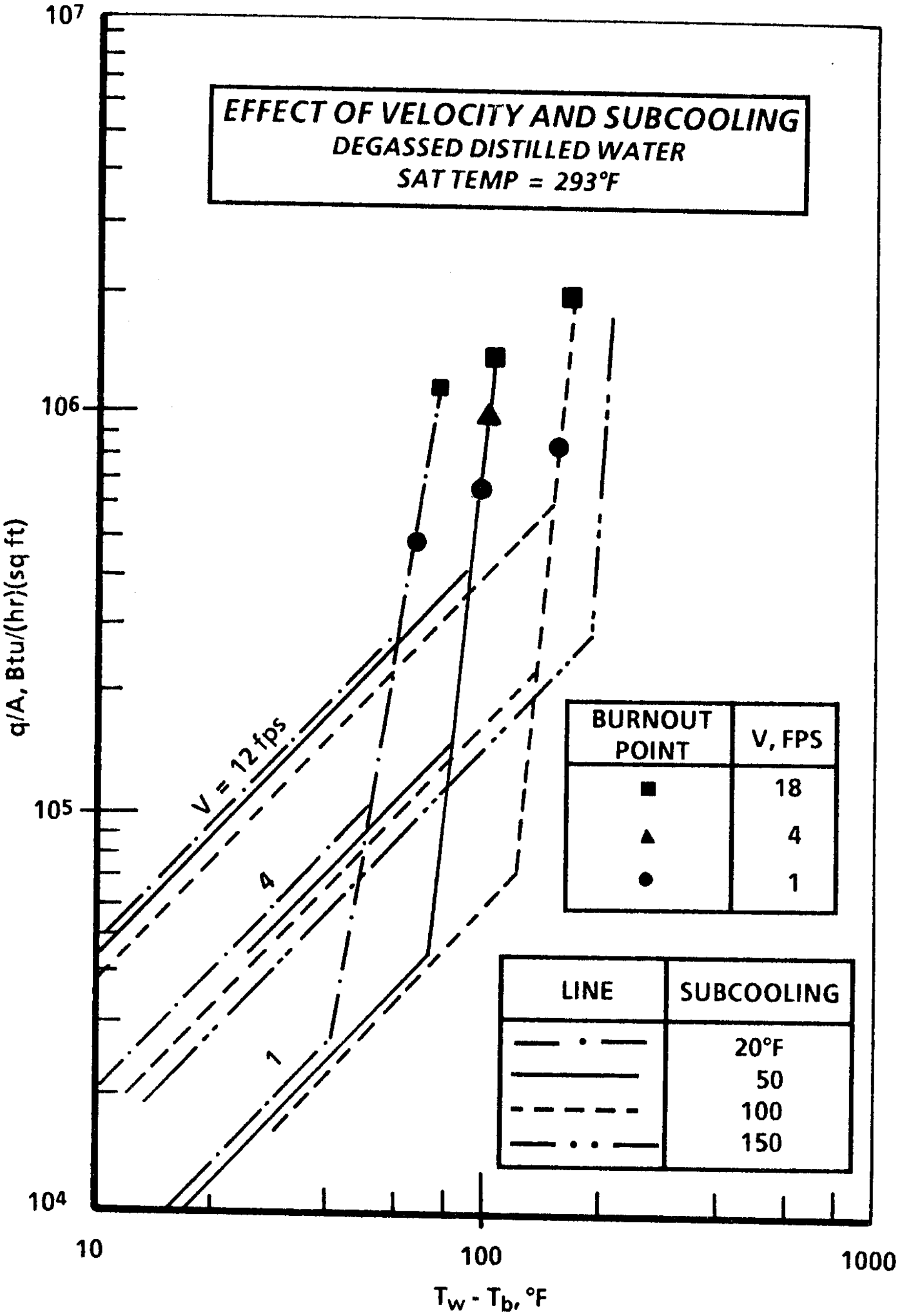


Figure 3. Effect of Velocity and Subcooling

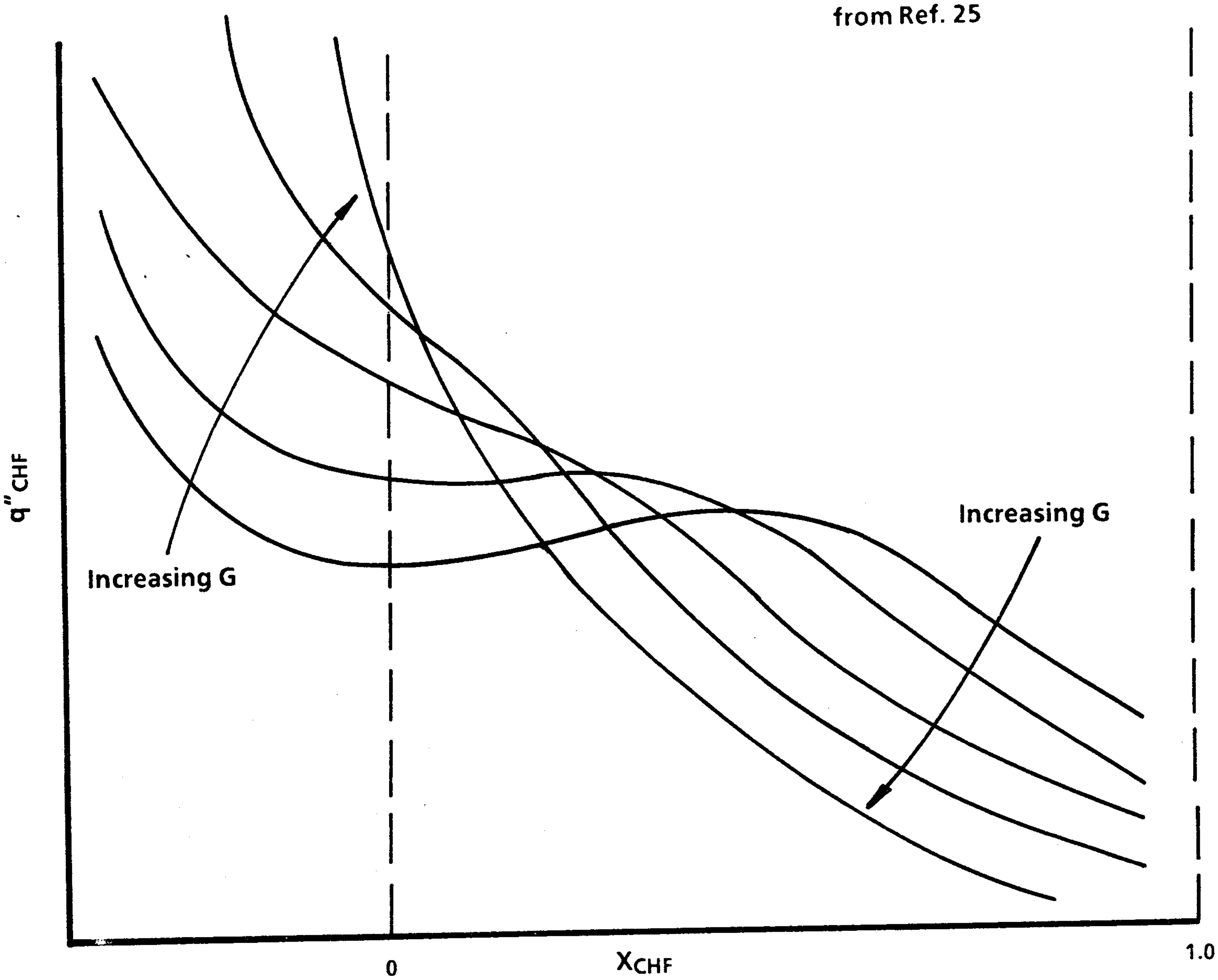


Figure 4. Effect of Mass Velocity and Vapor Quality

from Ref. 26

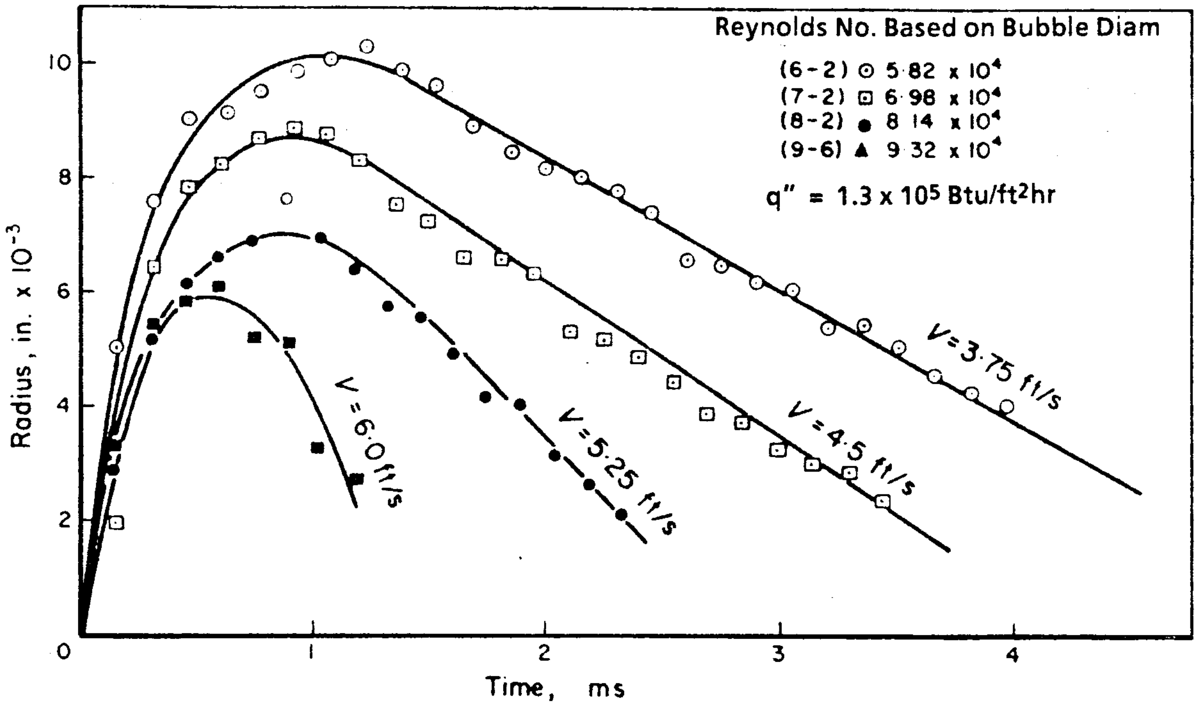


Figure 5. Effect of Liquid Velocity on Bubble Growth and Collapse

from Ref. 31

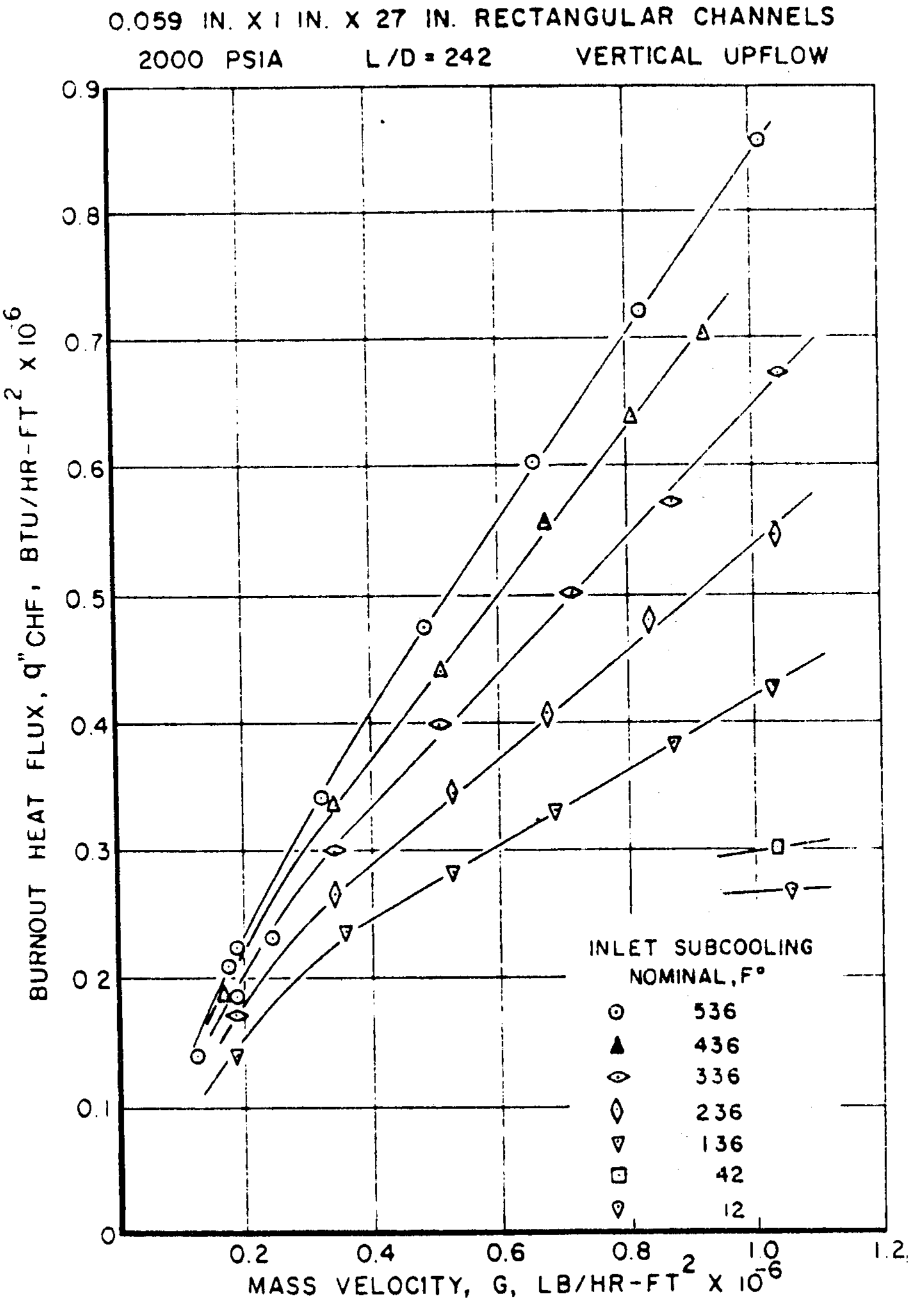


Figure 6. Effect of Subcooling with Increasing Mass Velocity

from Ref. 29

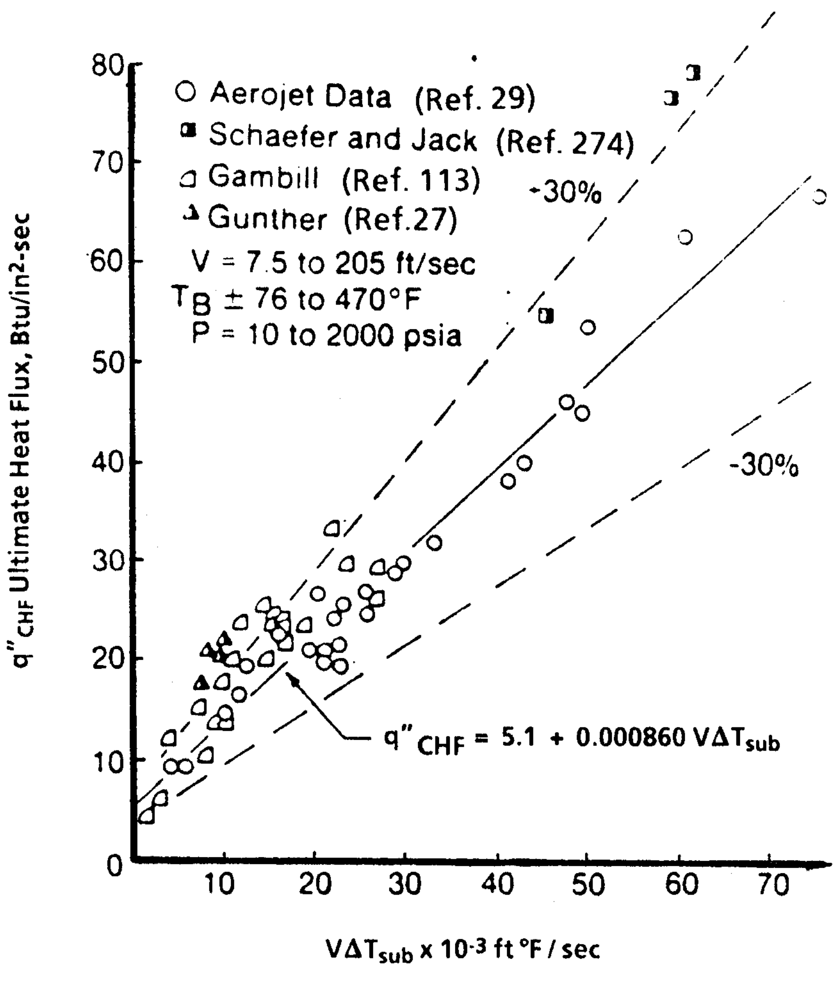


Figure 7. Subcritical Pressure Water Burnout Heat Flux

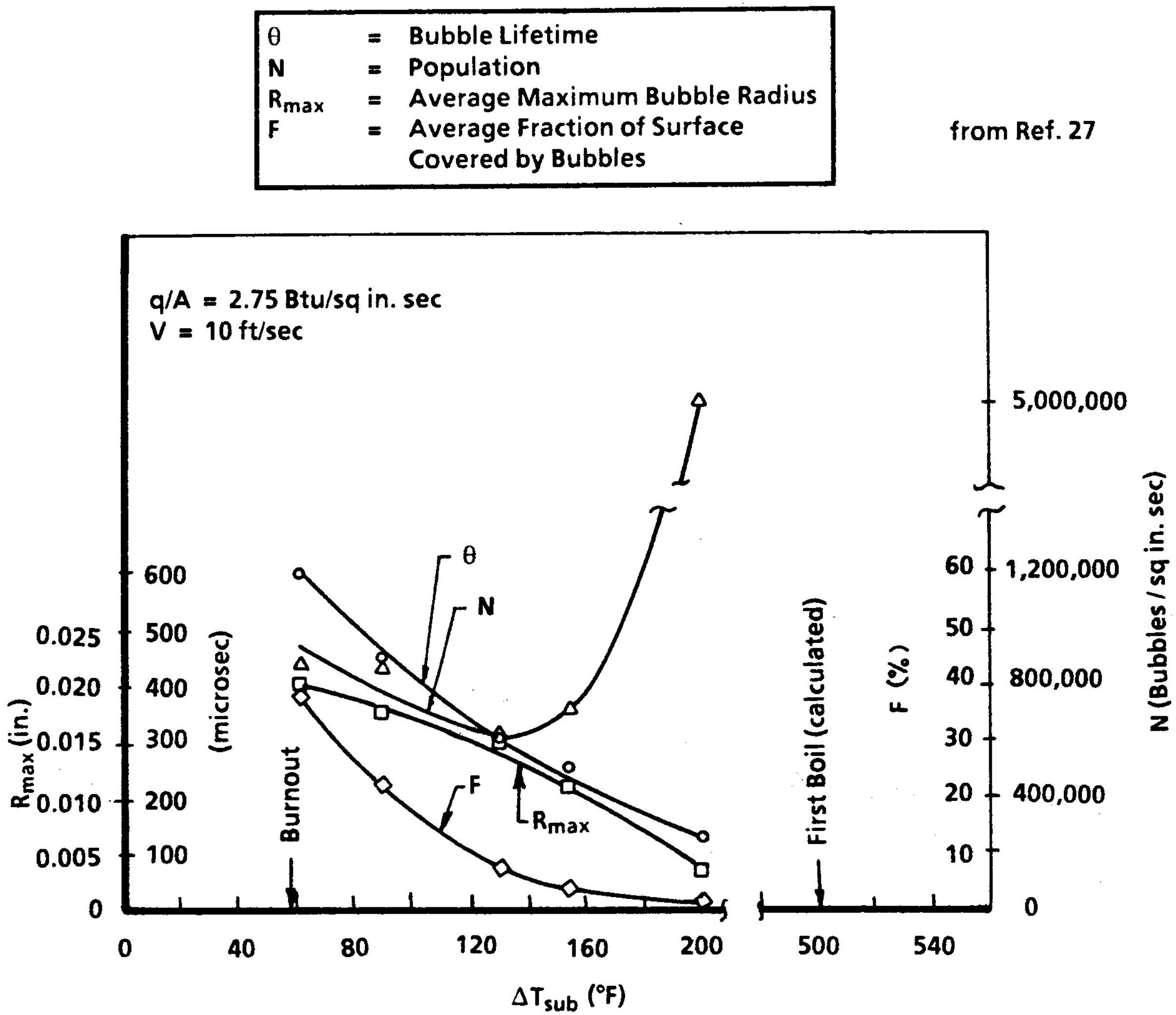


Figure 8. Effect of Subcooling on Bubble Characteristics

from Ref. 36

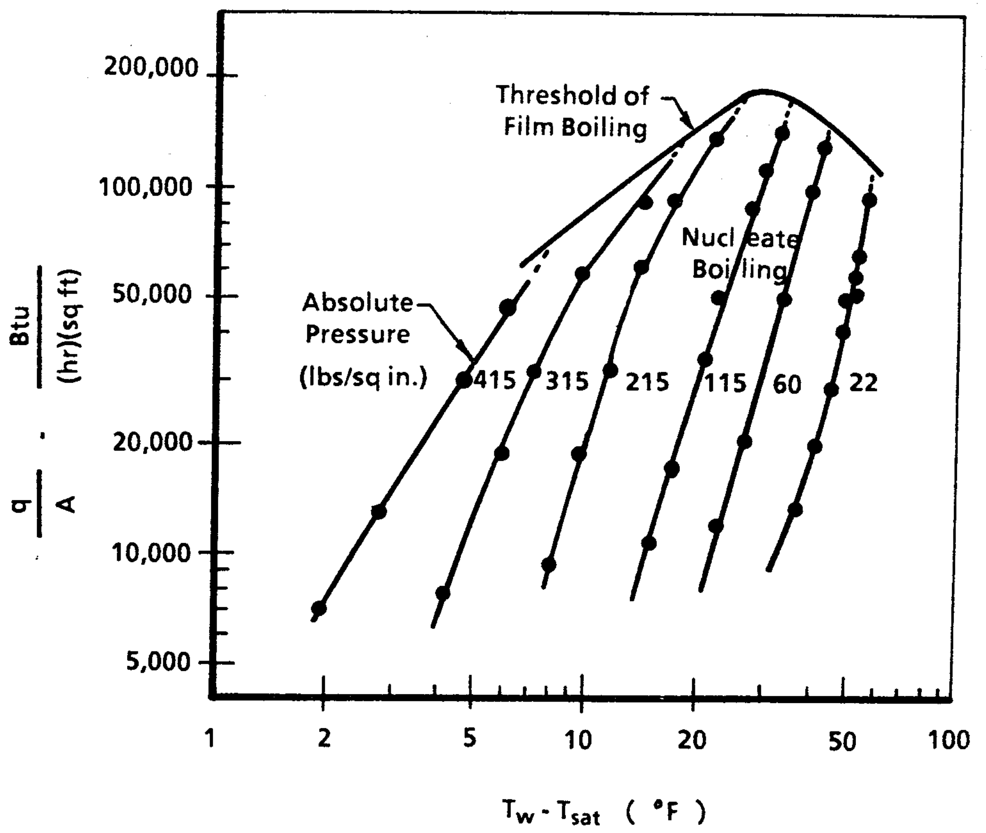


Figure 9. Optimum Pressure for the Boiling of n-Pentane

from Ref. 15

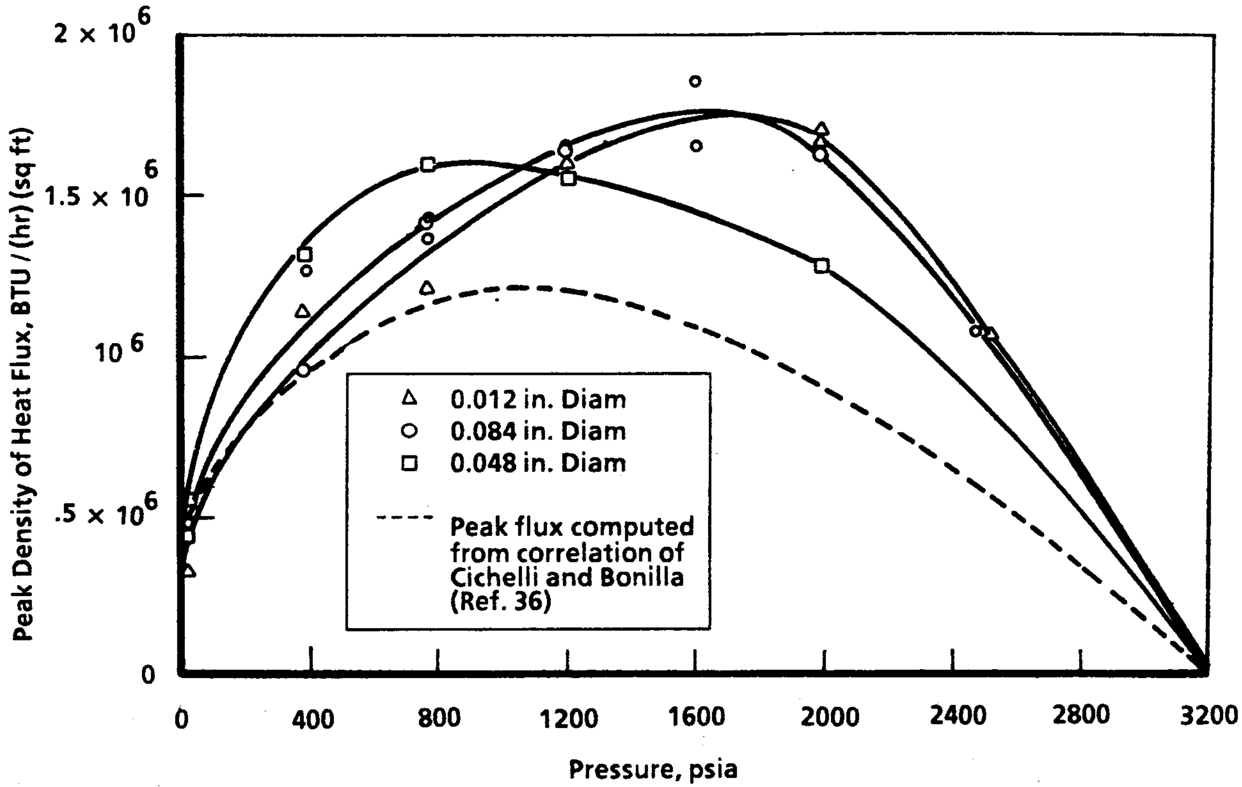


Figure 10. Effect of Heater Size on Optimum Pressure

from Ref. 1

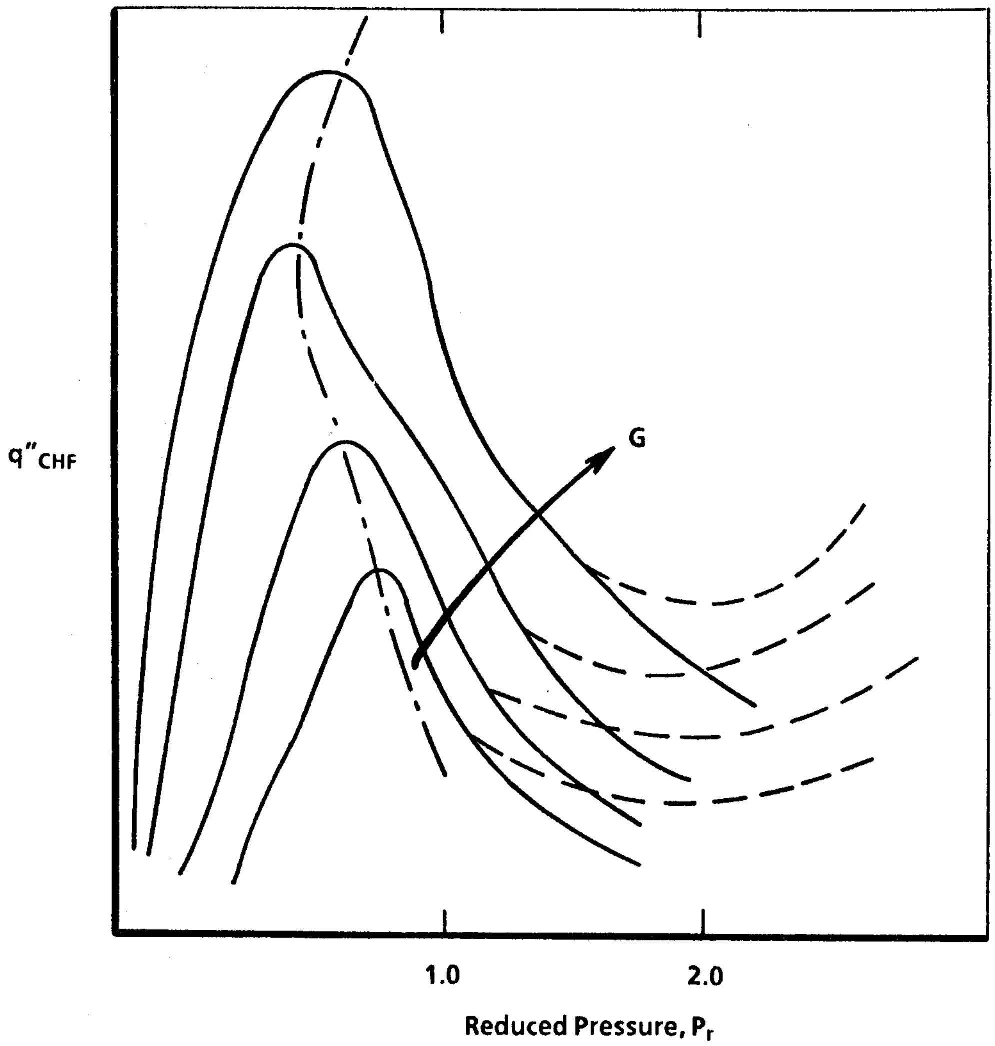


Figure 11. Effect of Mass Velocity on Optimum Pressure

from Ref. 37

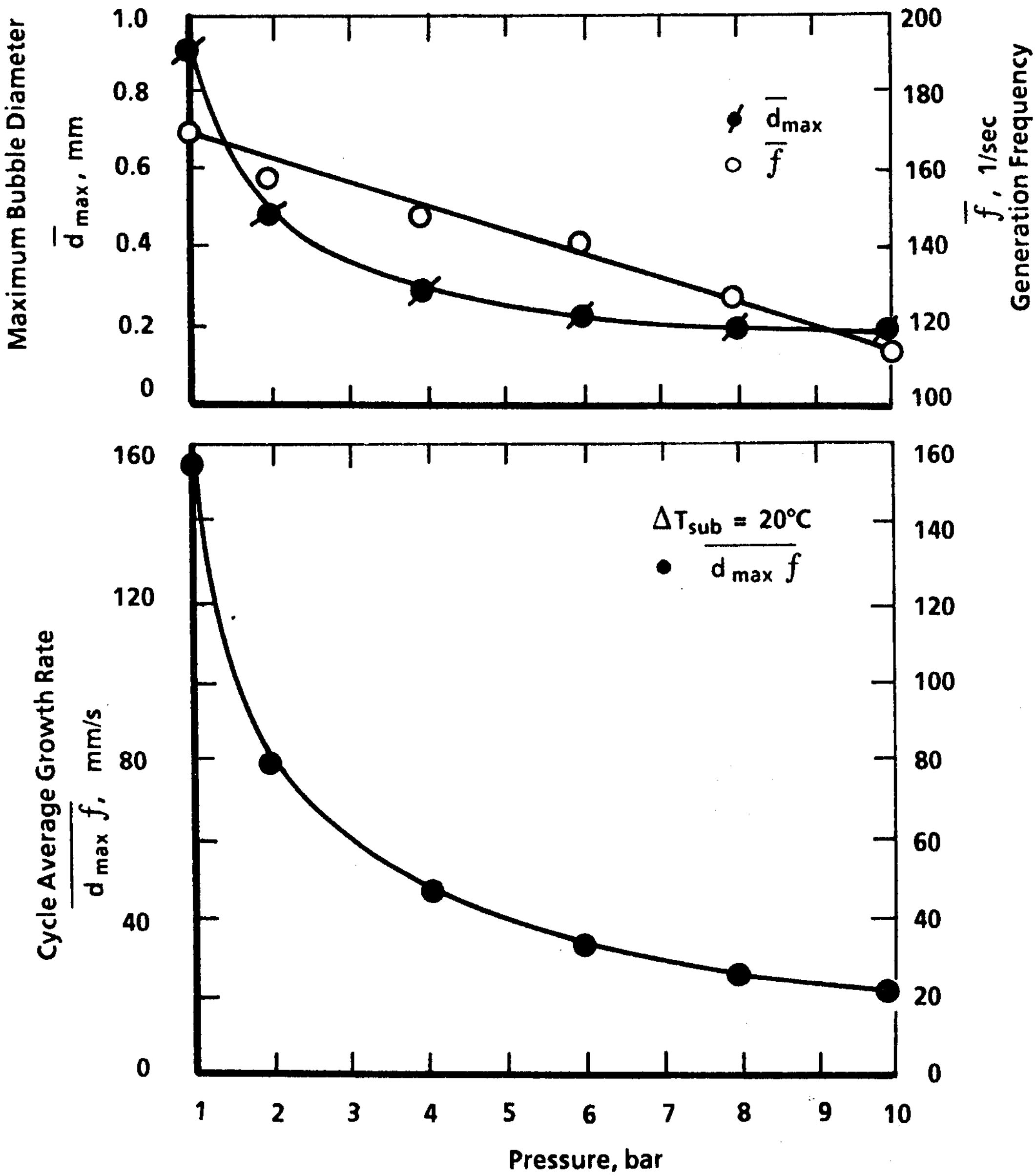


Figure 12. **Effect of Pressure on Bubble Characteristics**

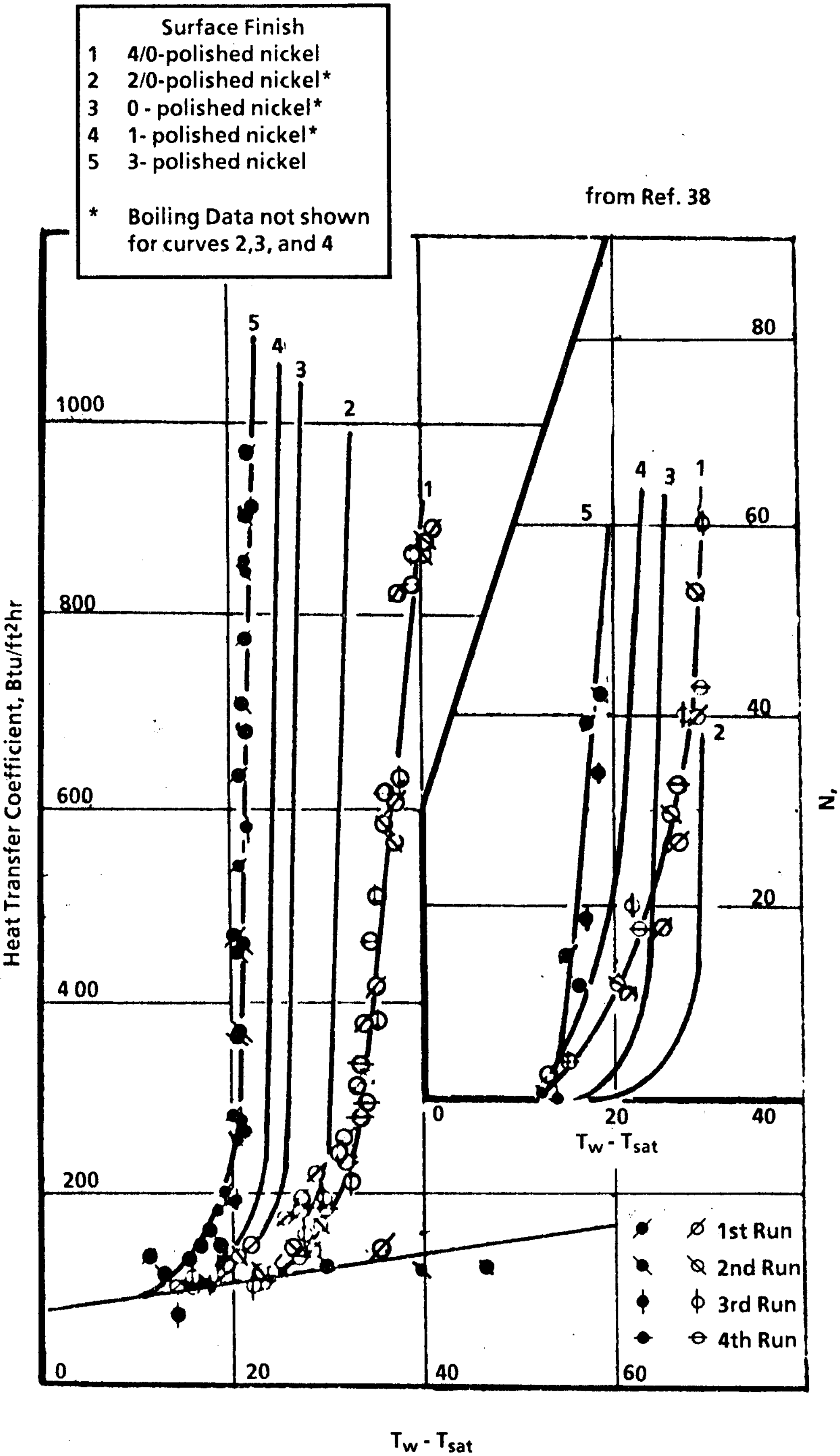


Figure 13. Effect of Surface Roughness on Heat Transfer

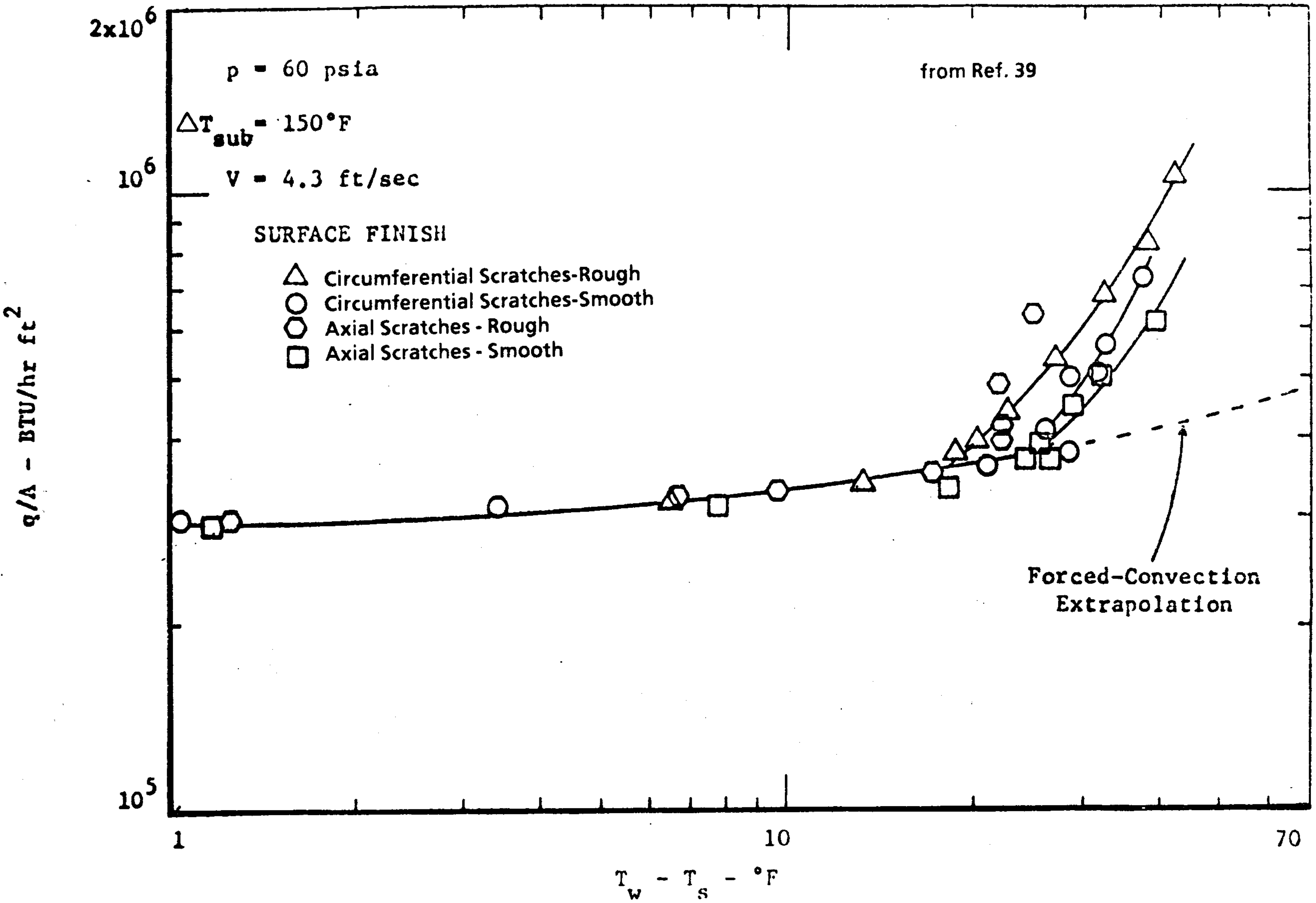


Figure 14. Roughness Effect on Heat Transfer

from Ref. 46

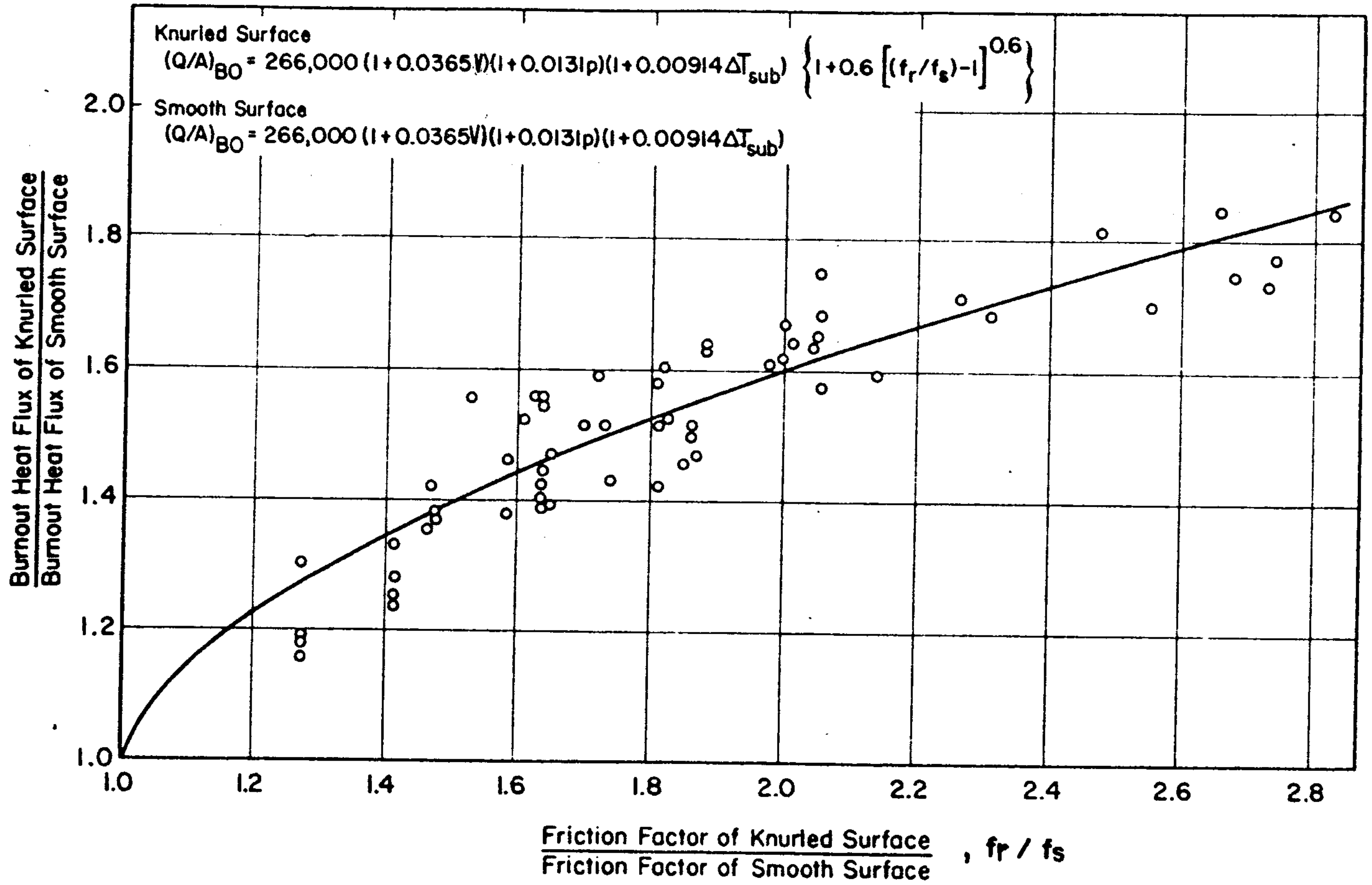


Figure 15. Burnout Heat Flux of Knurled Surfaces

from Ref. 53

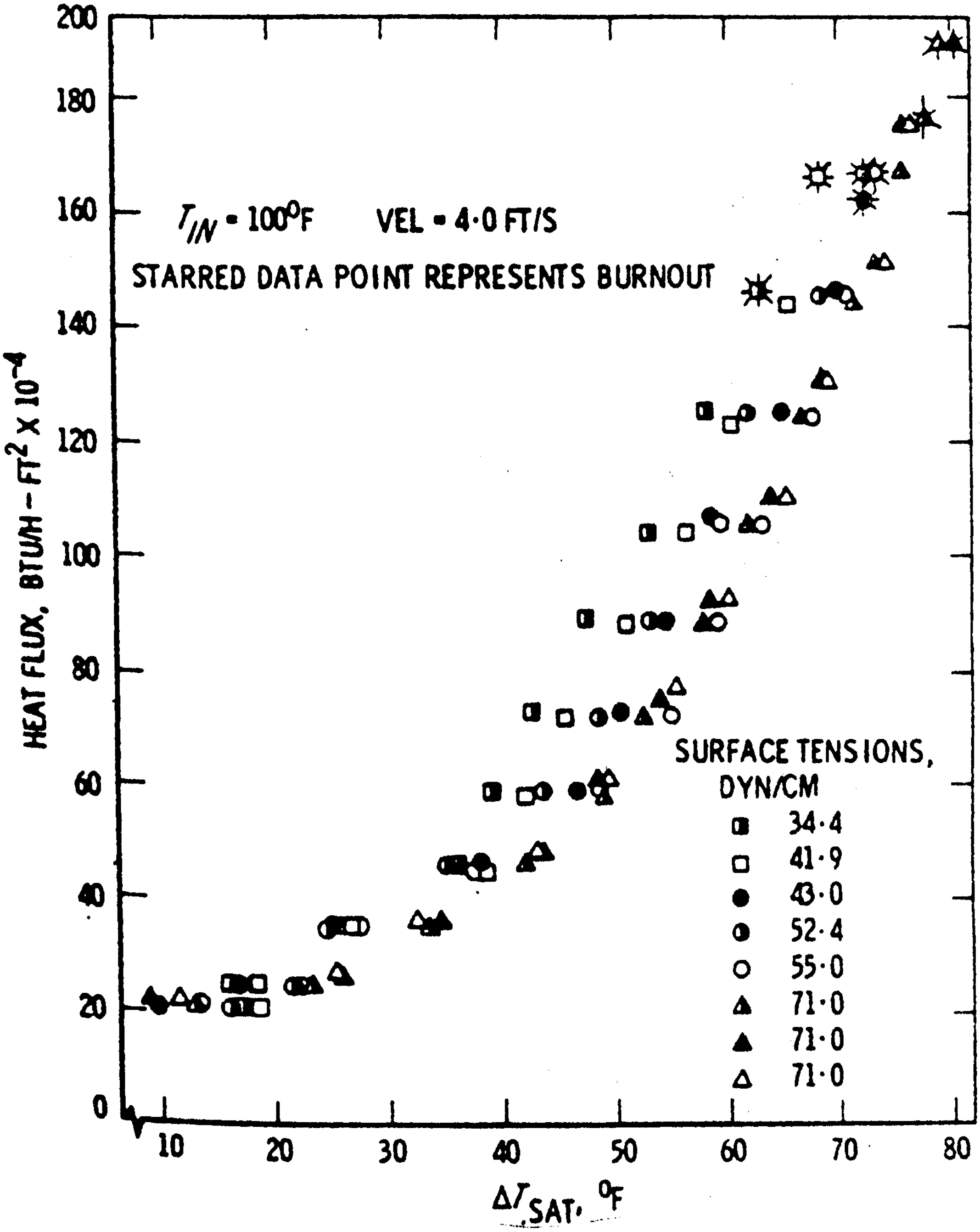
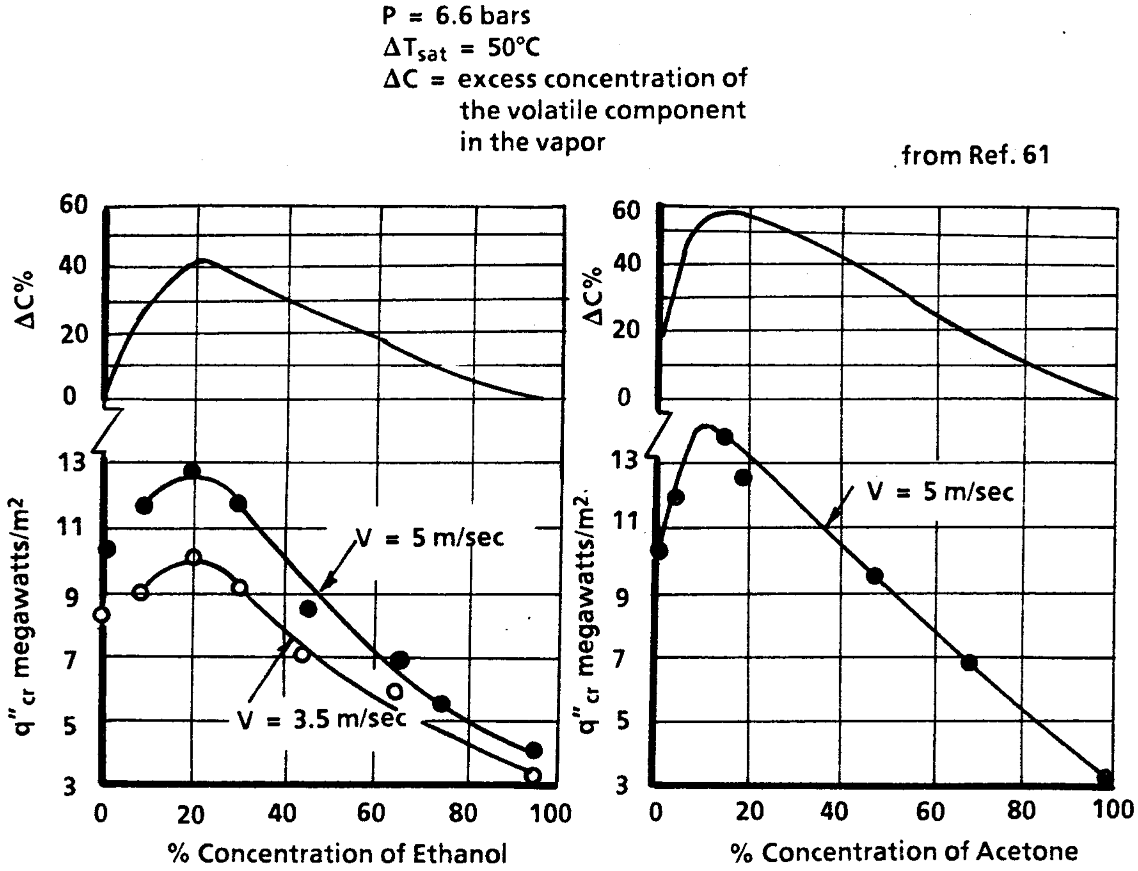


Figure 16. Effects of Surface Active Agents on Heat Transfer



a. Ethanol-Water

b. Acetone-Water

Figure 17. Effect of Binary Mixture Concentrations on CHF

from Ref. 74

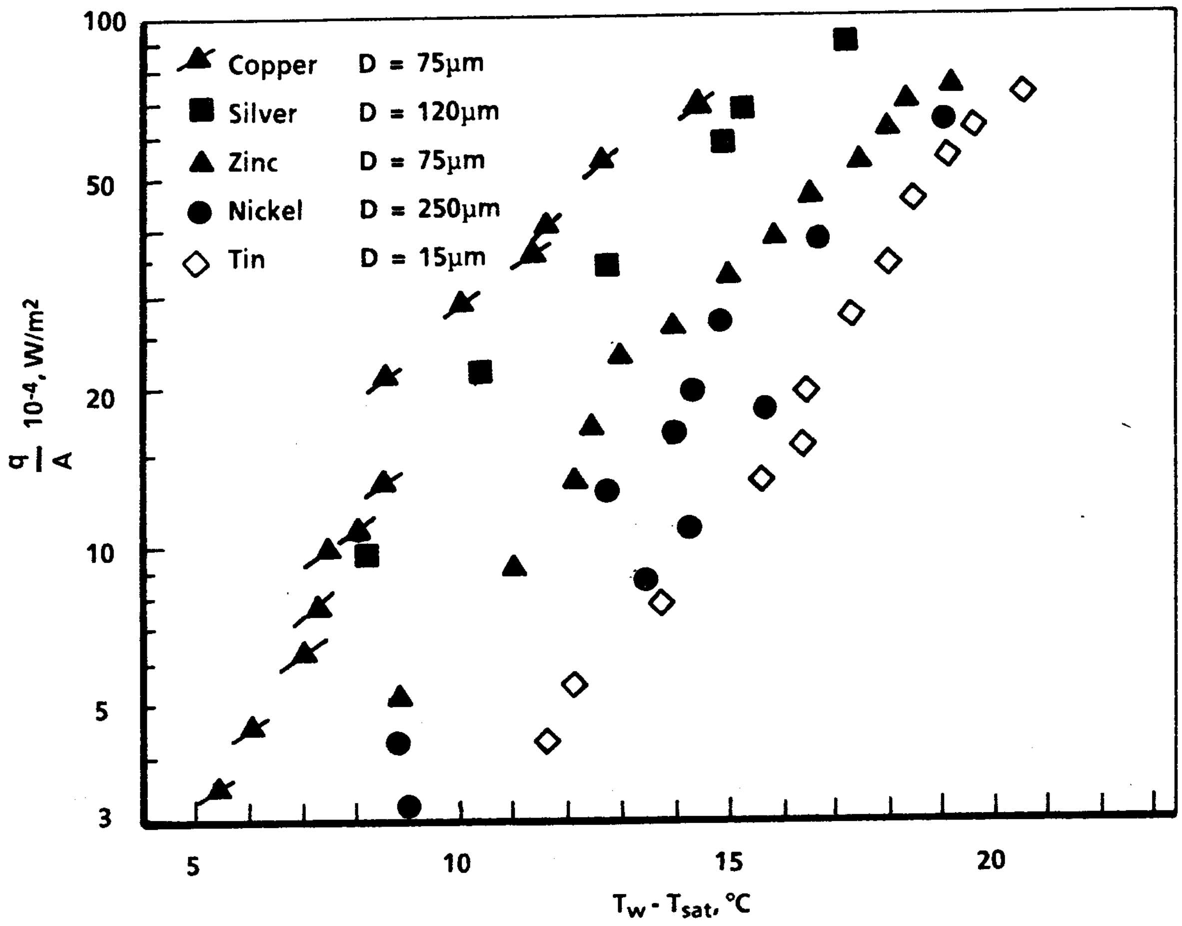


Figure 18. Effect of Wall Material on Heat Transfer

from Ref. 76

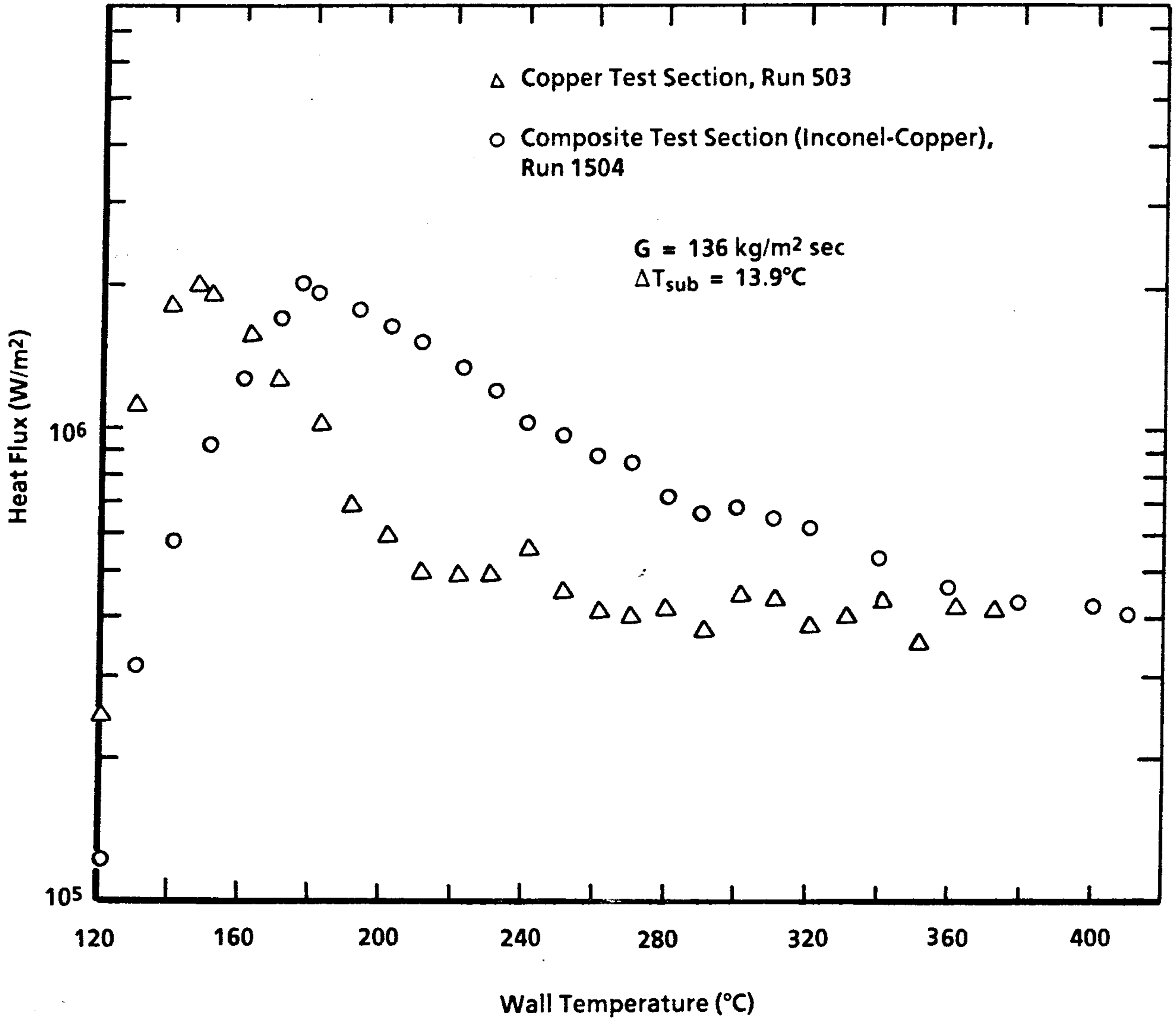


Figure 19. Effect of Wall Material on CHF

from Ref. 79

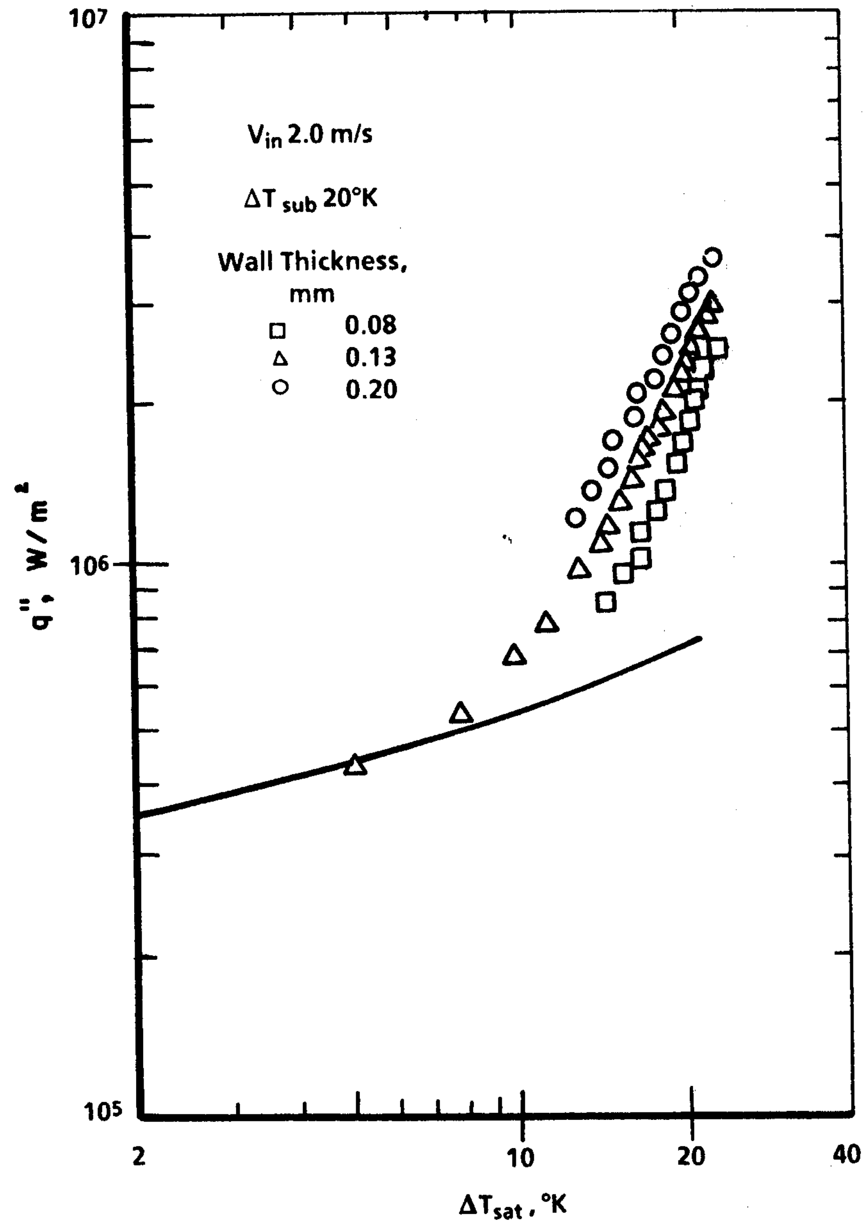


Figure 20. Effect of Wall Thickness on Heat Transfer

from Ref. 7

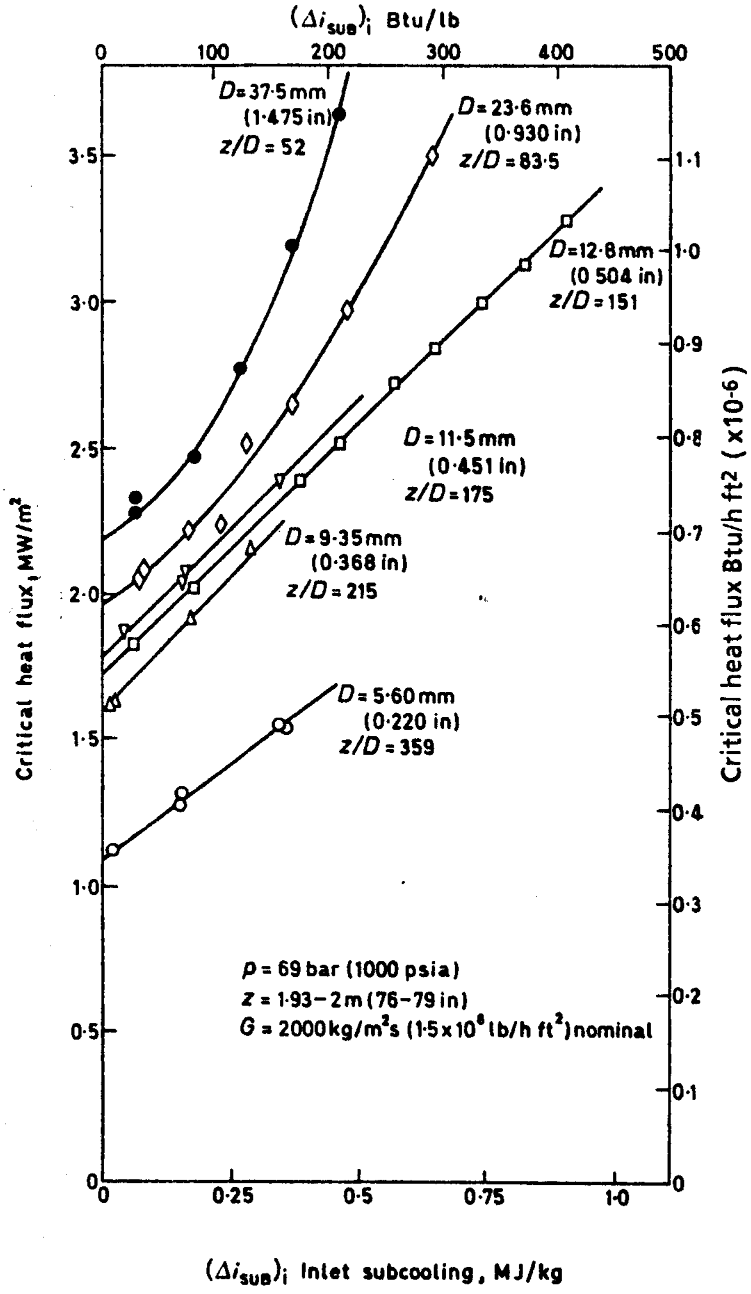


Figure 21. The Influence of Tube Diameter on CHF at Various Subcoolings

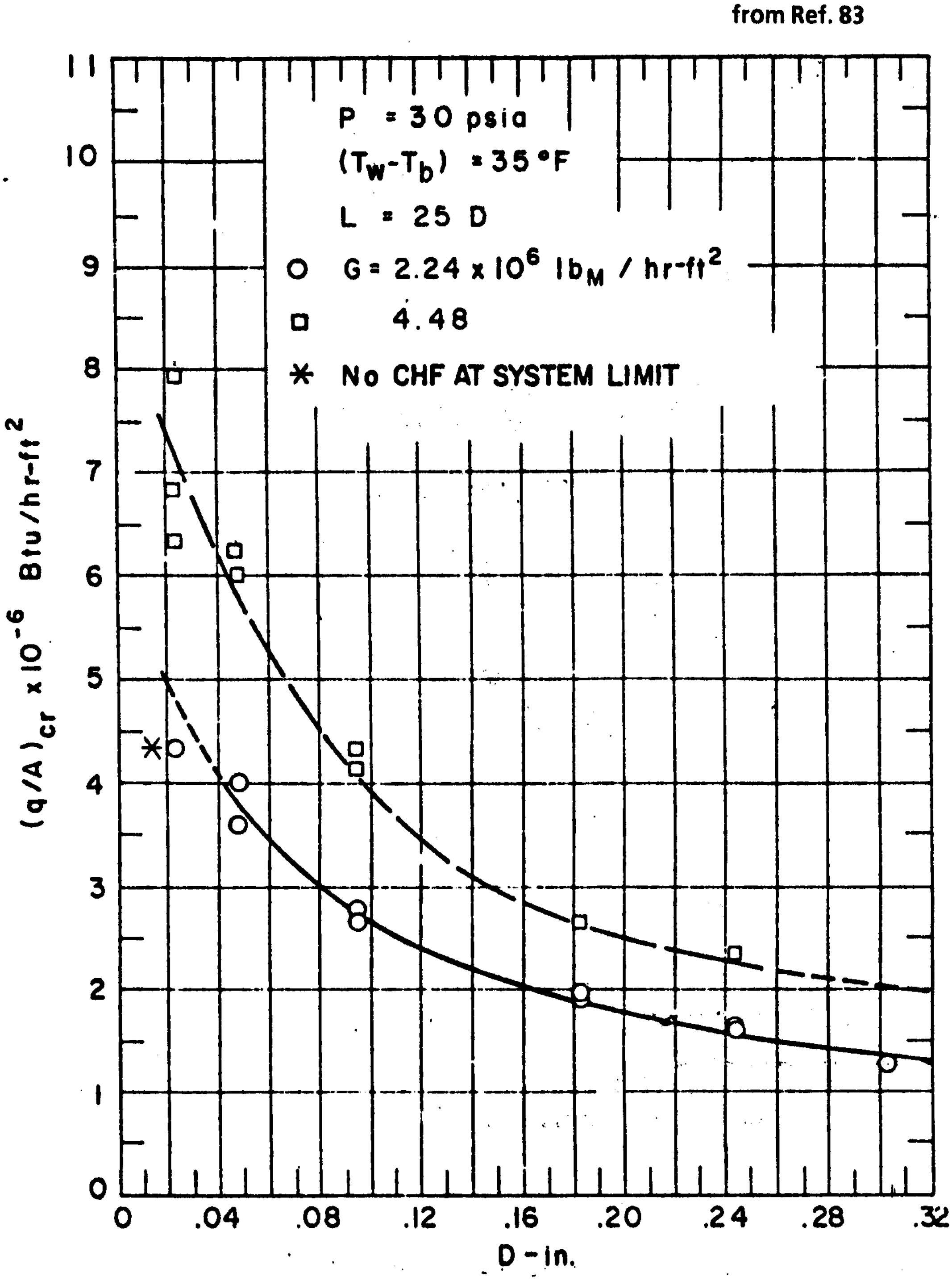


Figure 22. Effect of Tube Diameter on CHF

from Ref. 83

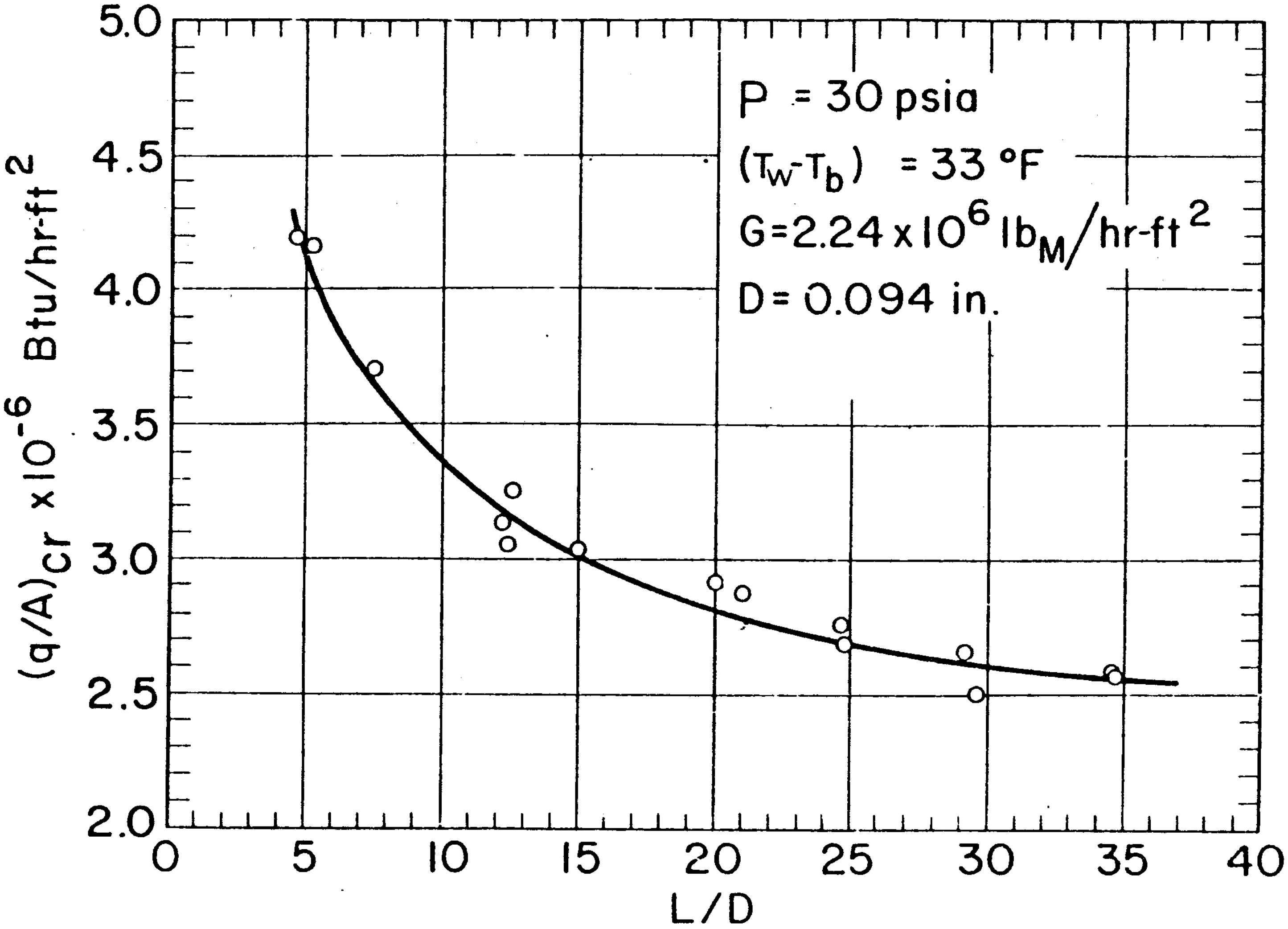


Figure 23. Effect of Heated Length on CHF

from Ref. 7

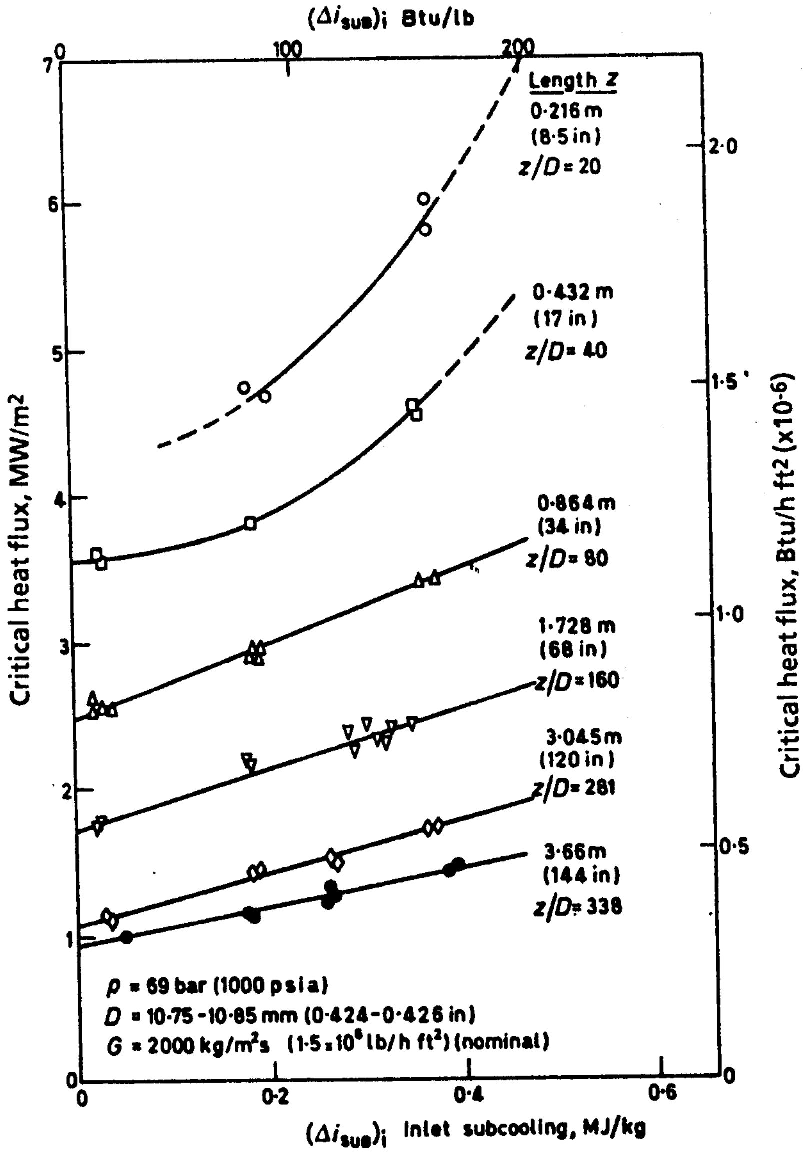
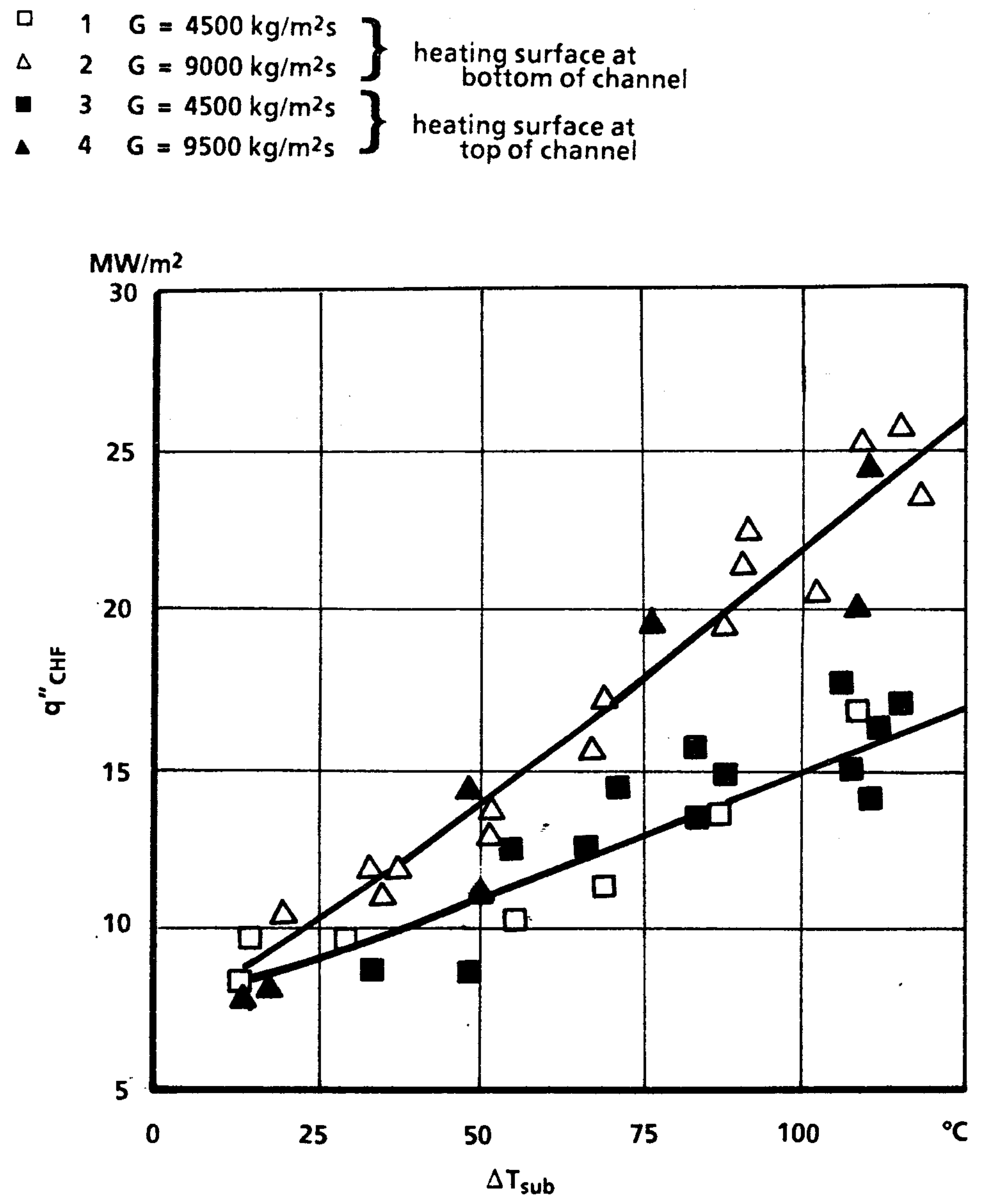


Figure 24. Effect of Tube Length on CHF at Various Subcoolings



from Ref. 93

Figure 25. Effect of Position of the Heating Surface on CHF

from Ref. 24

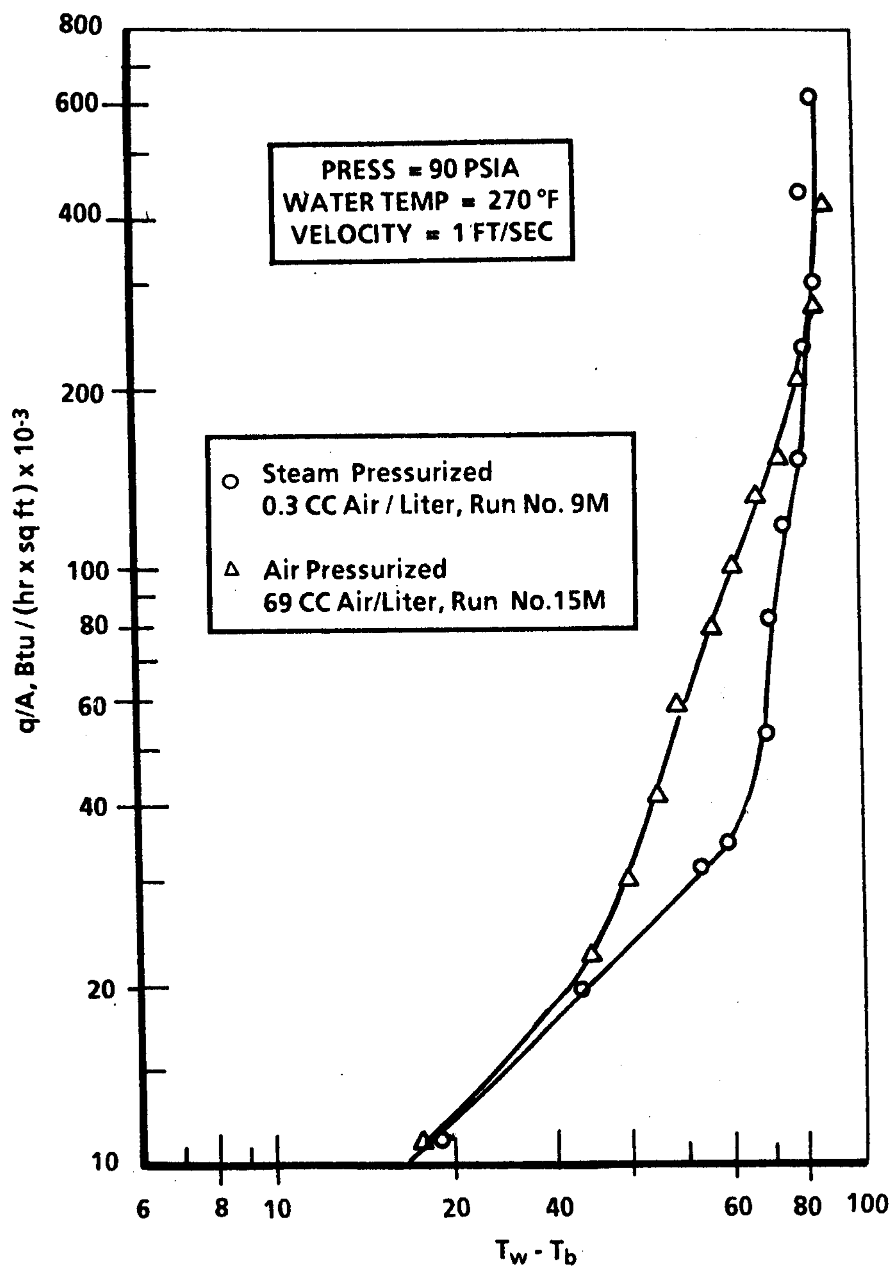


Figure 26. Effect of Gas Content on Heat Transfer

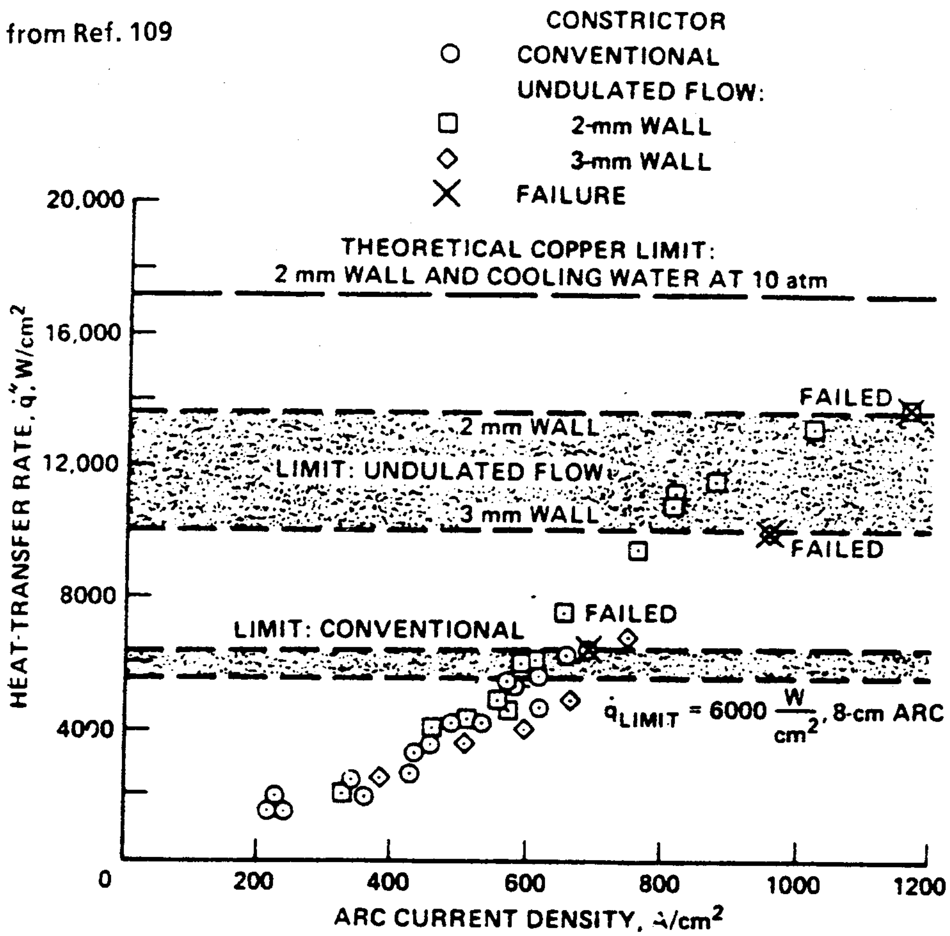
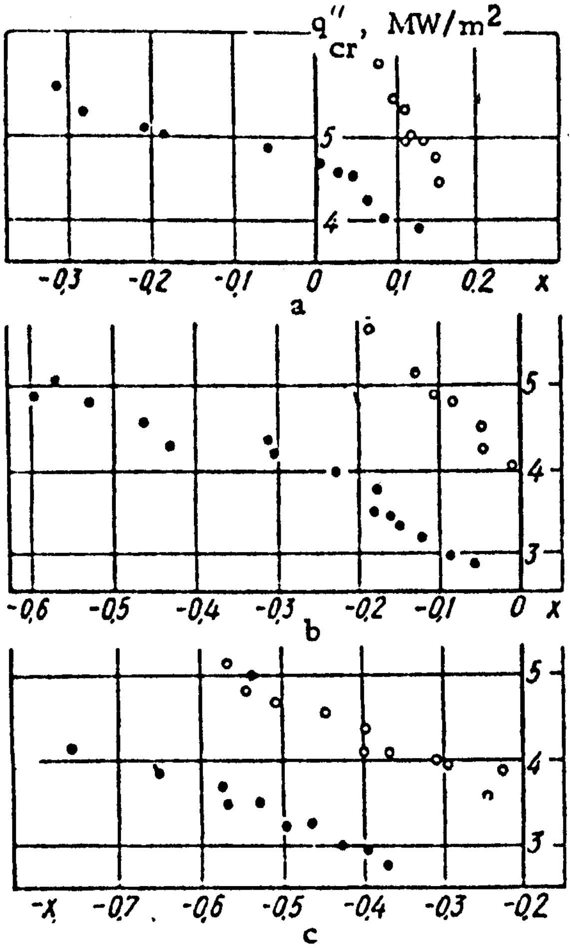


Figure 28. Effect of Undulating Flow on Heat Transfer

from Ref. 112



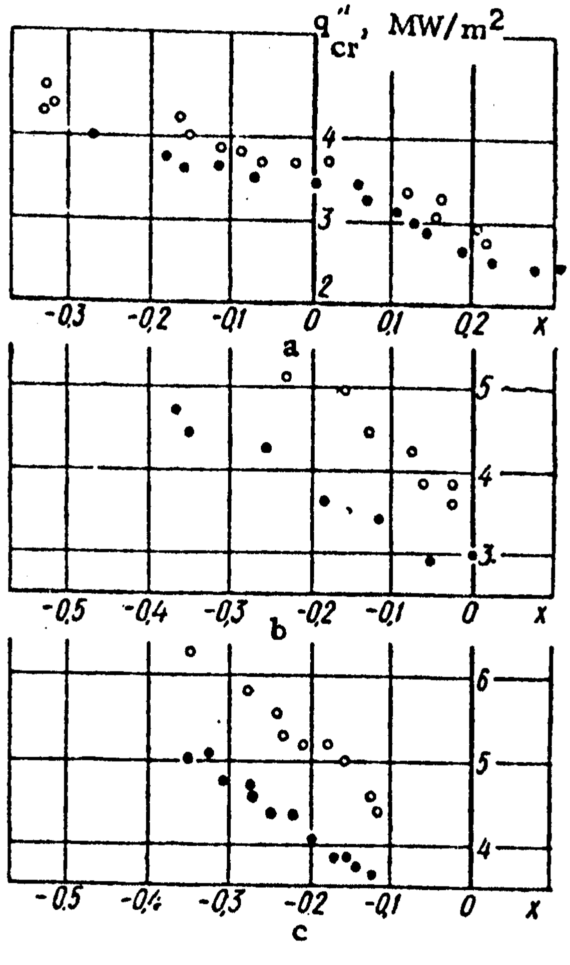
a = 98.1 bar
b = 147.2 bar
c = 186.5 bar

solid symbols - no swirl
open symbols - with swirl

G = 1700 kg/m-sec
L = 270 mm

Figure 29. Pressure Effect with Flow Swirl

from Ref. 112



a. G = 500kg/m2-sec
b. G = 1000kg/m2-sec
c. G = 1700kg/m2-sec

solid symbols - no swirl
open symbols - with swirl

p = 147.2 bar
L = 160 mm

Figure 30. Mass Velocity Effect with Flow Swirl

from Ref. 87

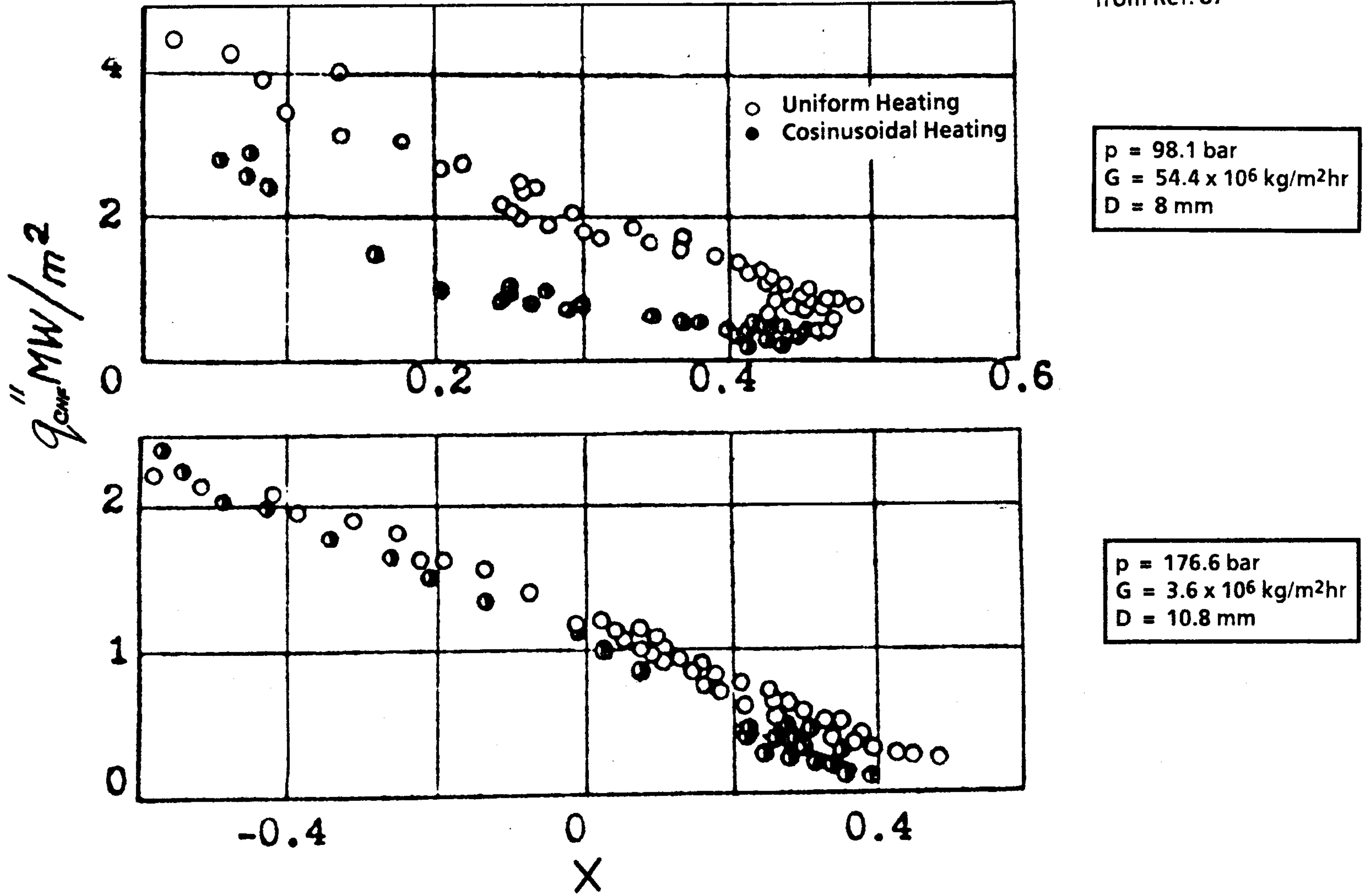


Figure 31. Effect of Non-Uniform Heating on CHF

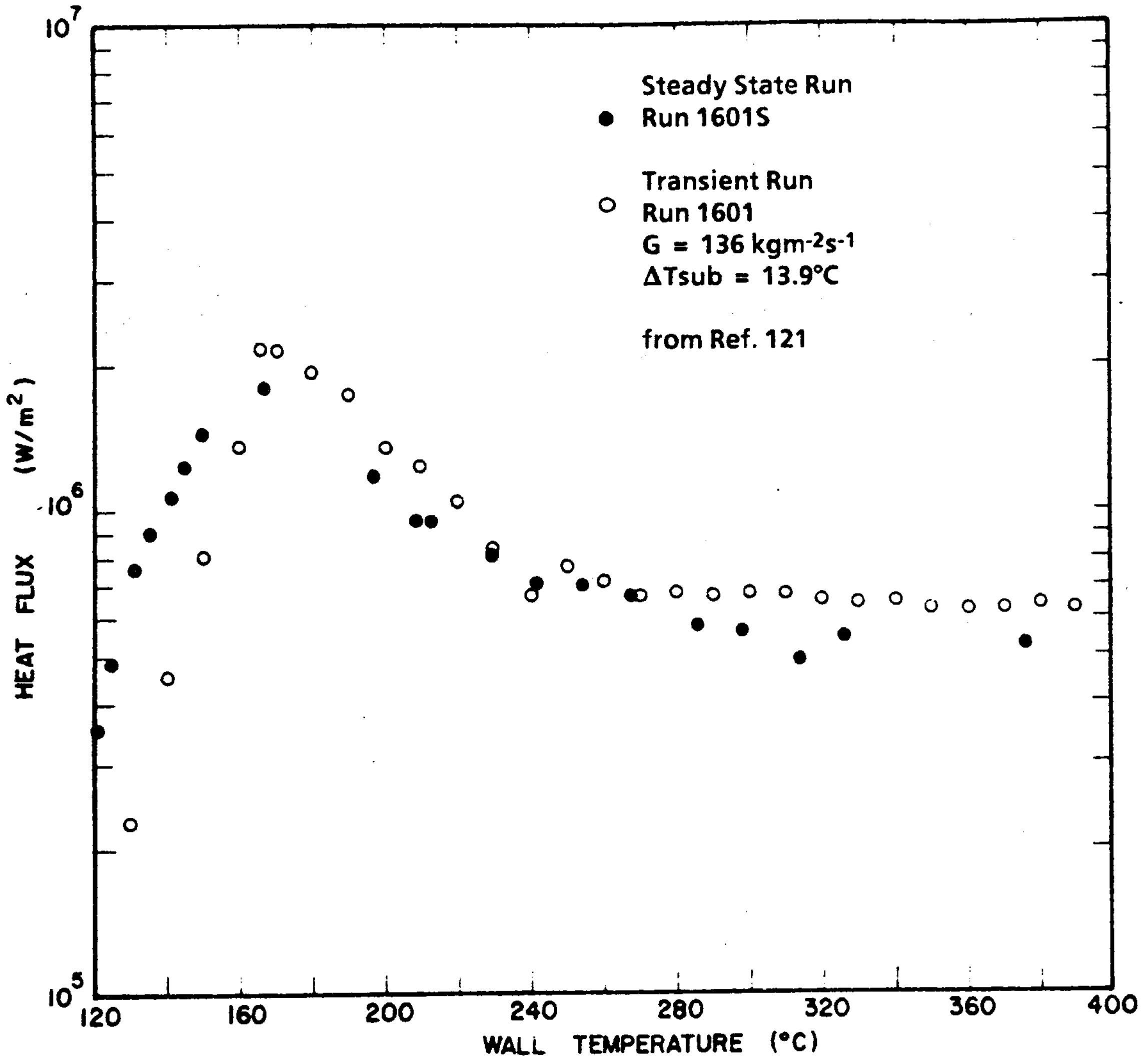
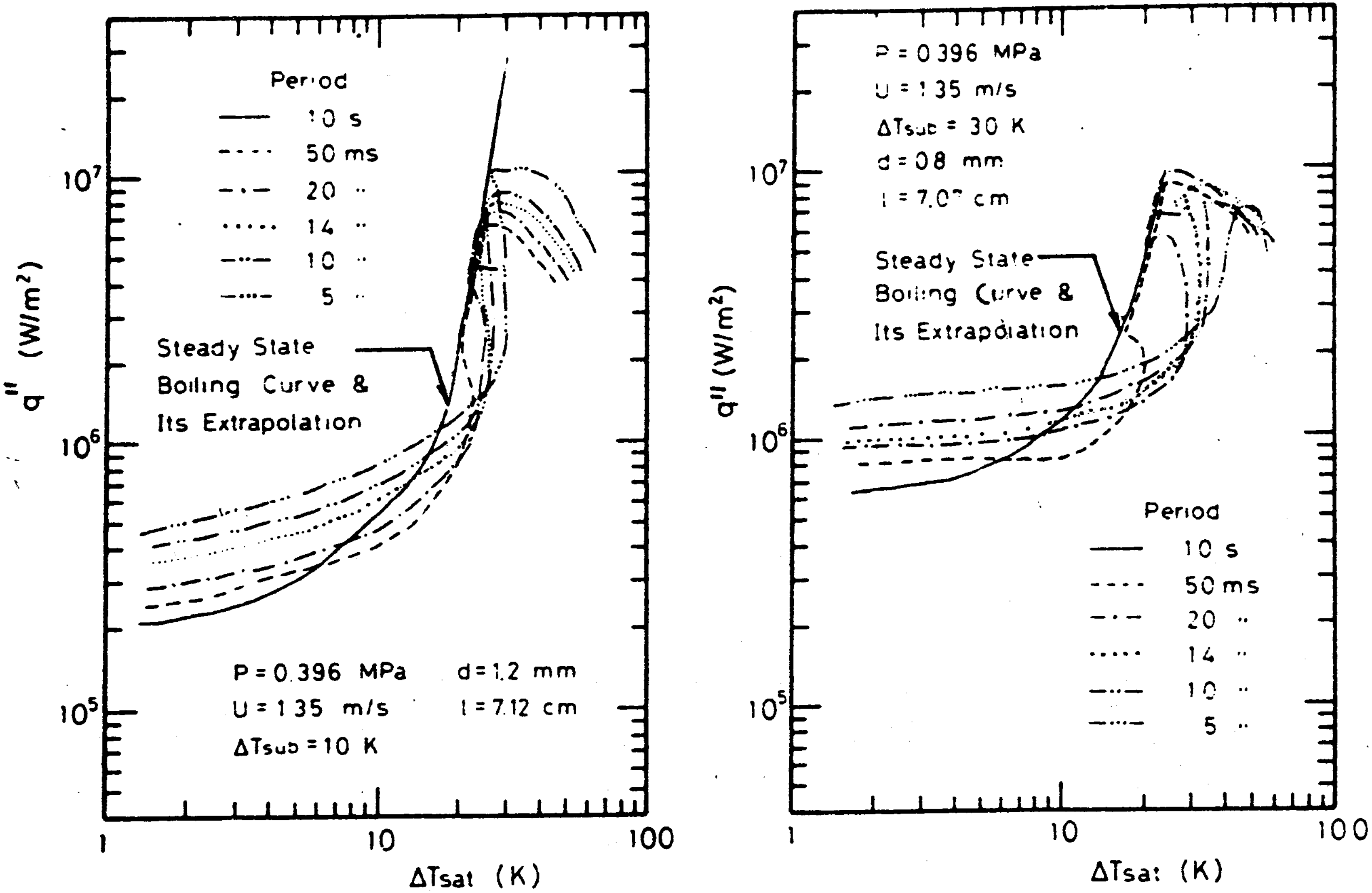


Figure 32. Comparison of Steady-State and Transient Heating Data

from Ref. 123



a. d = 1.2 mm and ΔT_{sub} = 10K

b. d = 0.8 mm and ΔT_{sub} = 30K

Figure 33. Effect of Transient Heating on Boiling Curve

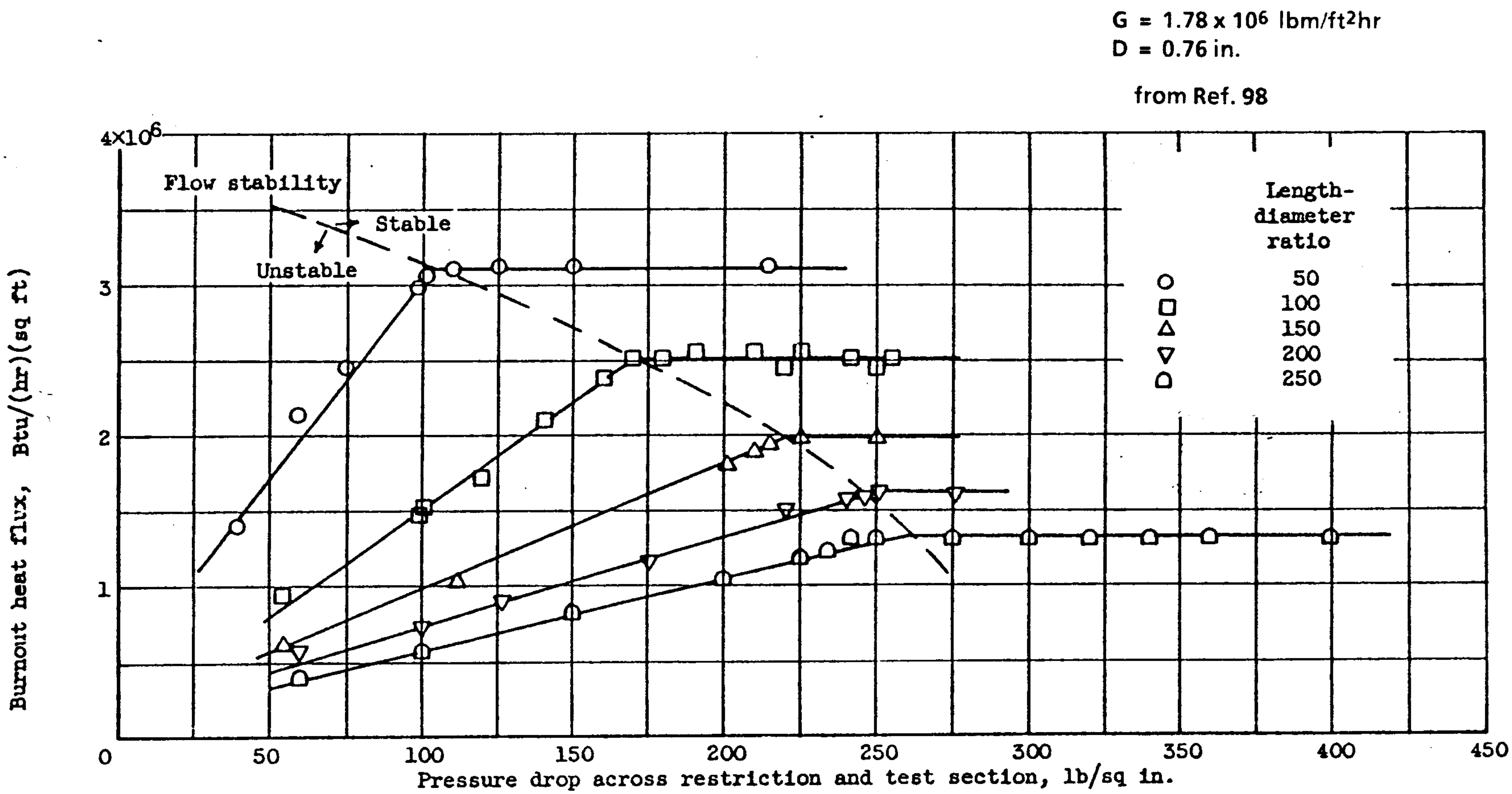


Figure 34. Effect of Flow Restriction on Flow Stability and Burnout

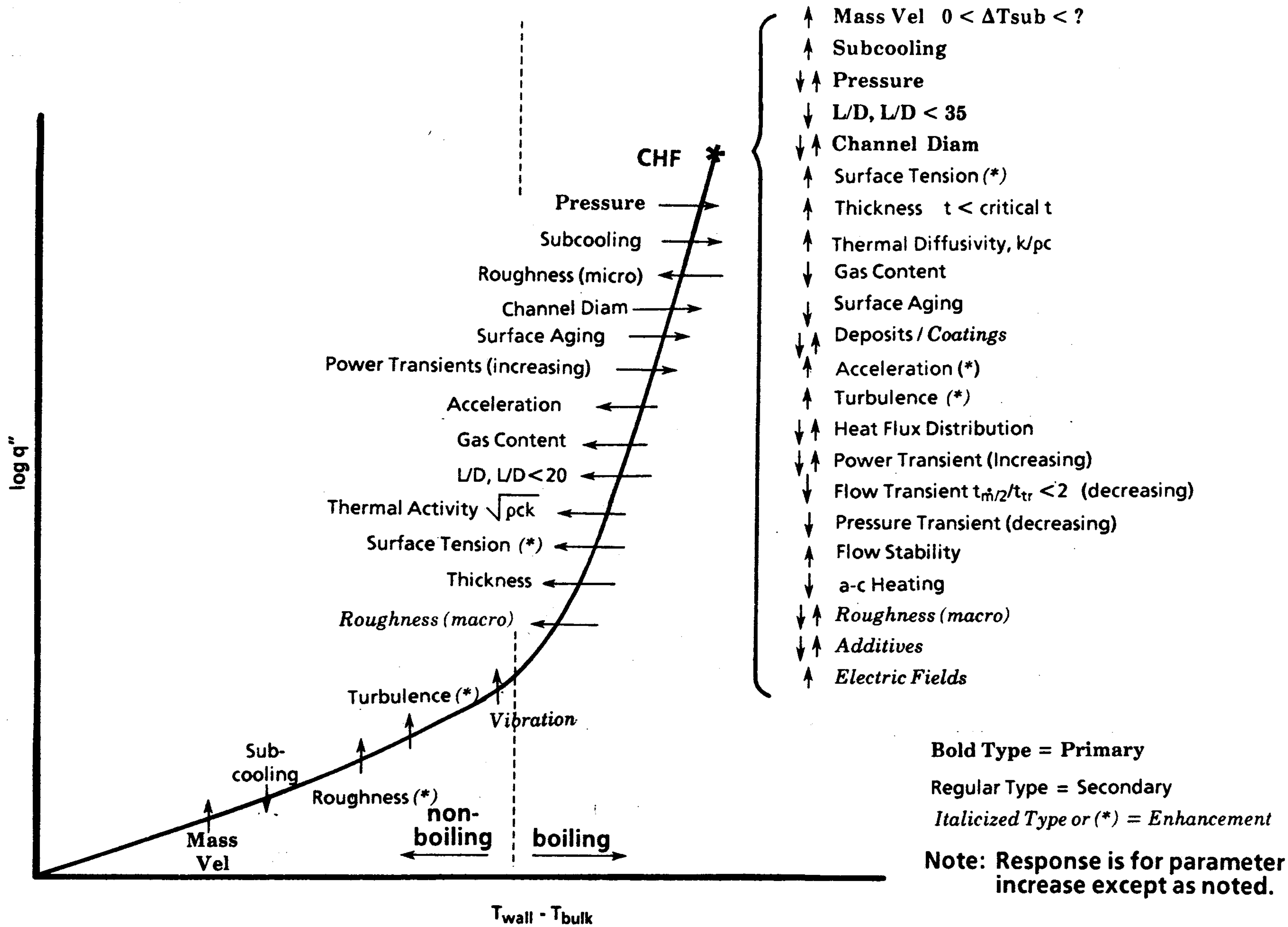


Figure 35. Parametric Trends for Subcooled, Forced-Convection Cooling

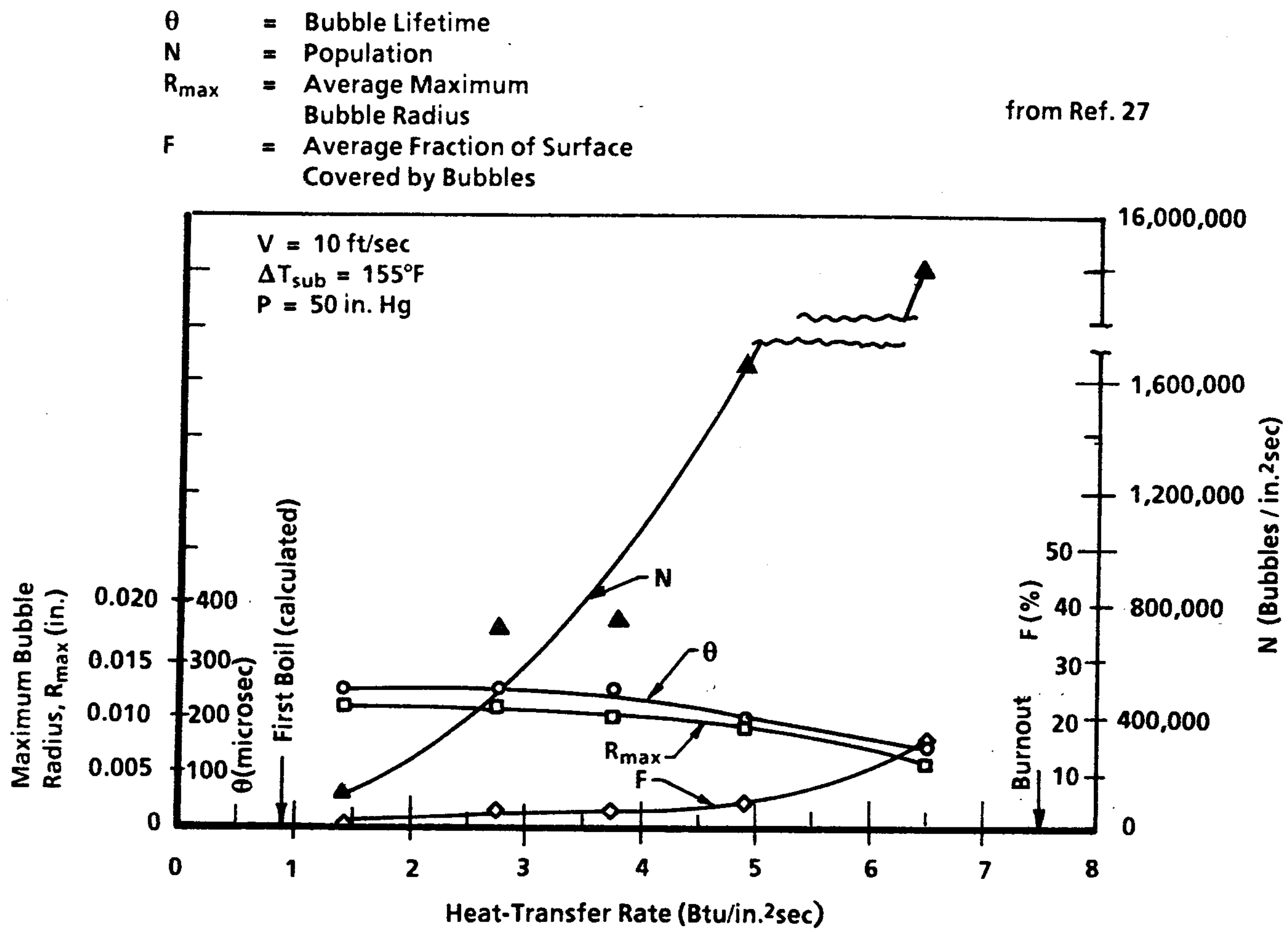


Figure 36. Effect of Heat-Transfer Rate on Bubble Characteristics

from Ref. 10

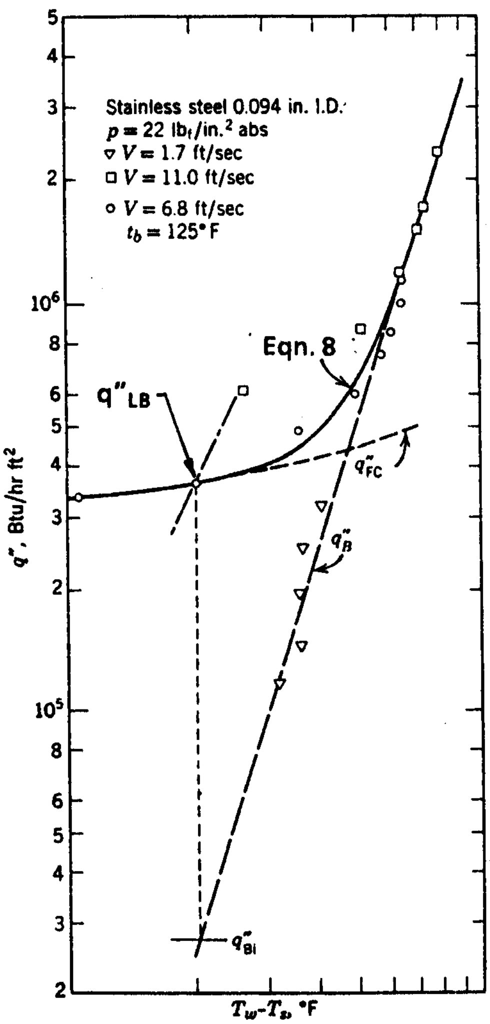


Figure 37. Construction of the Partial Boiling Curve

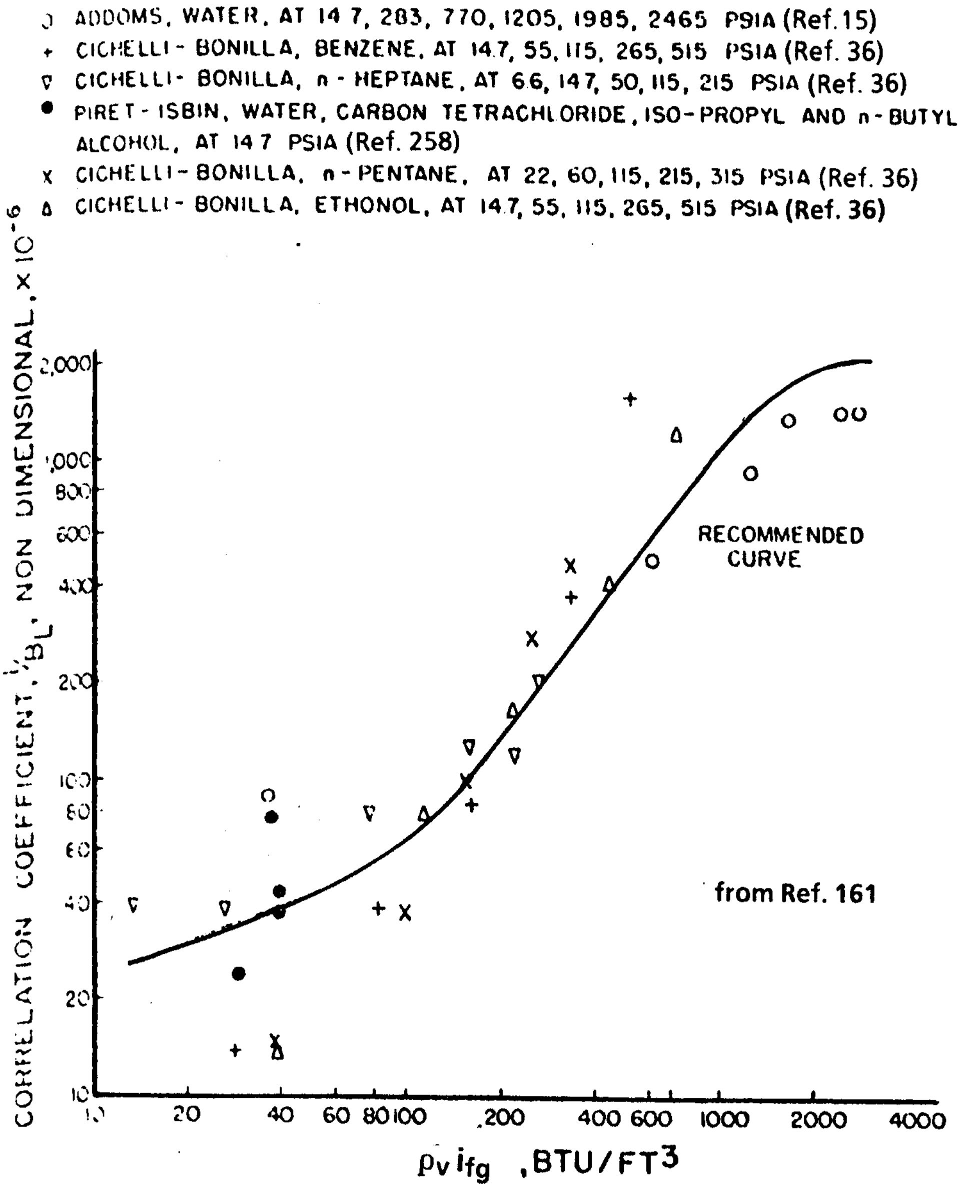


Figure 38. Determination of the Correlation B_L in the Levy Correlation

from Ref. 171

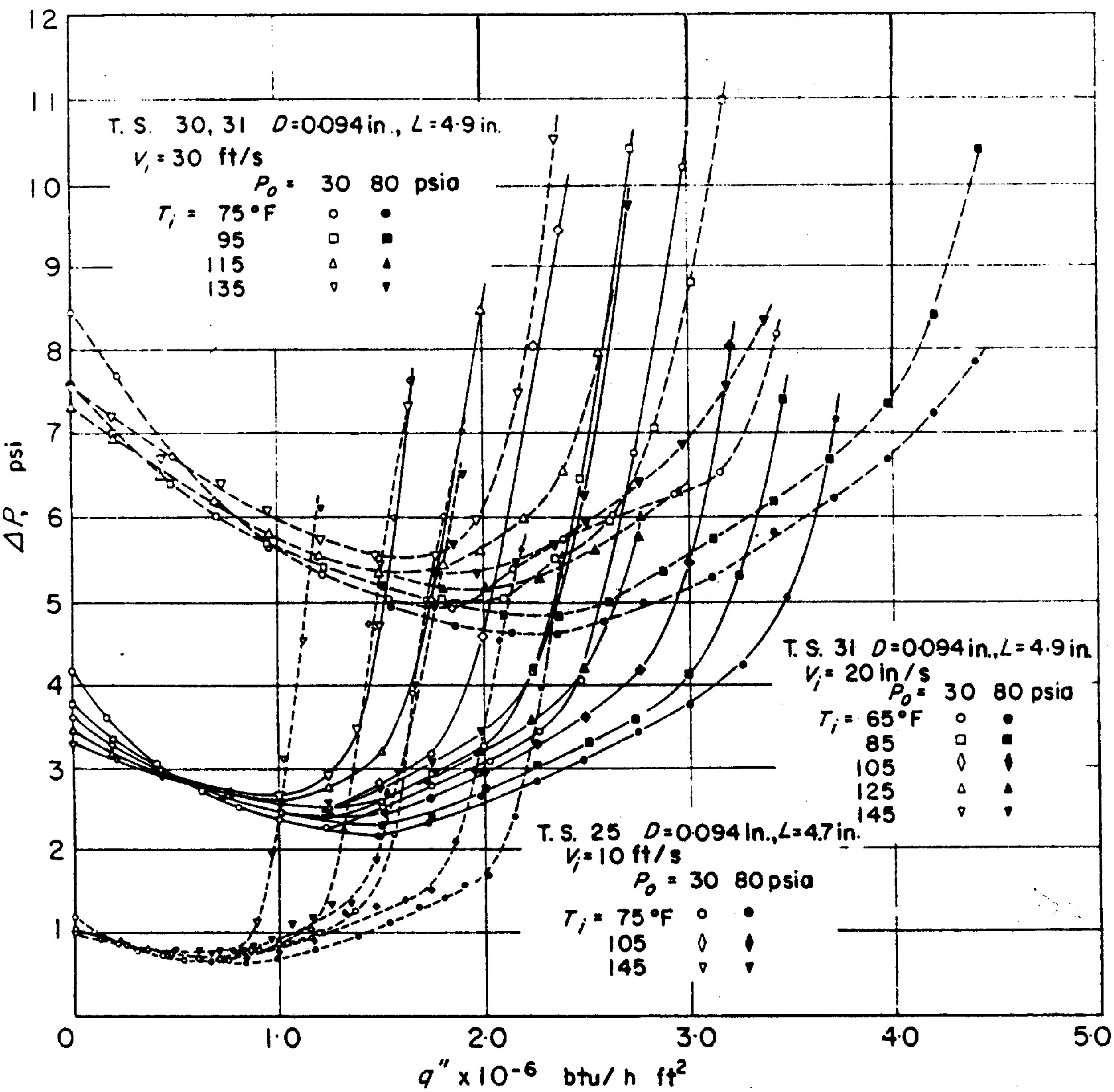


Figure 39. Dependence of Overall Pressure on Operating Conditions

from Ref. 202

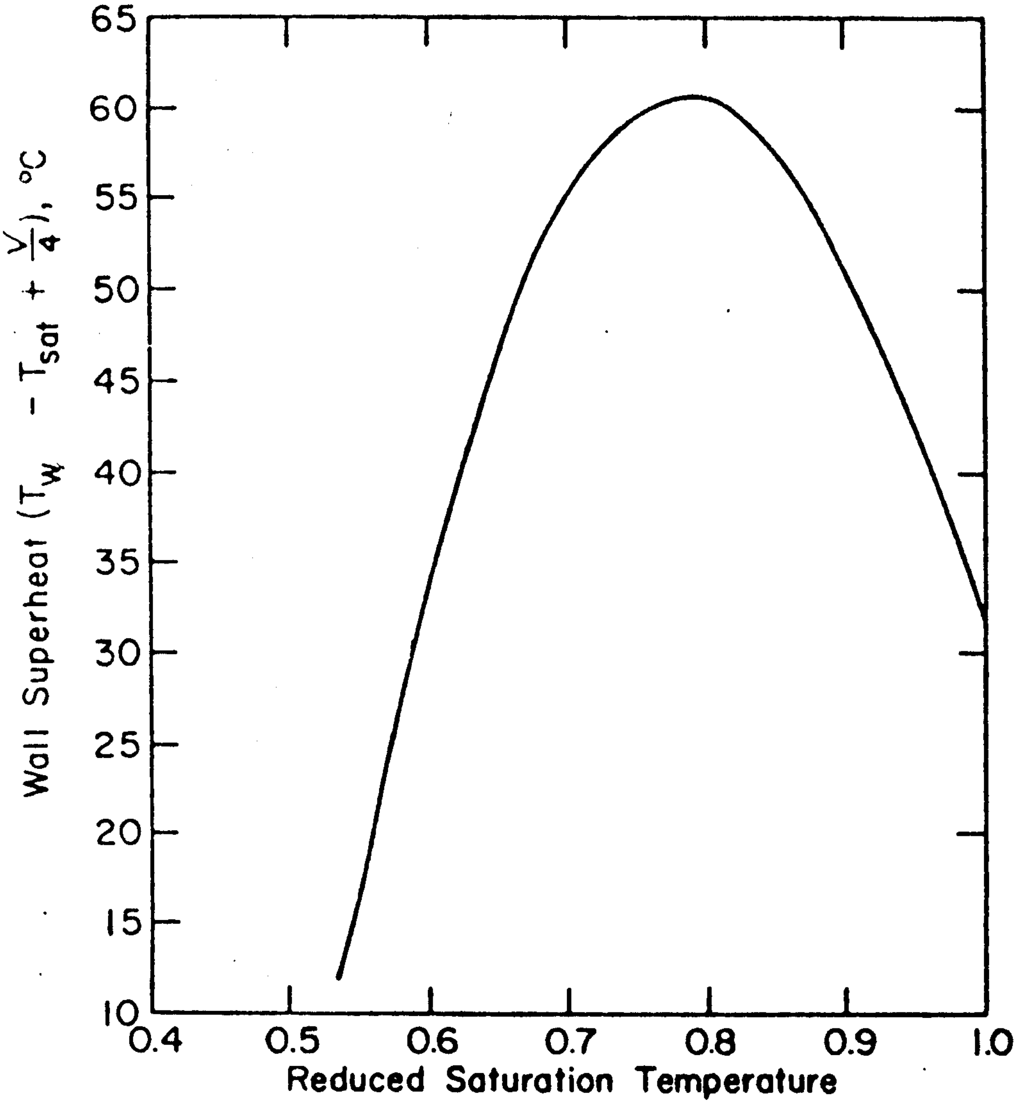


Figure 40. Generalized Wall Superheat at Burnout

from Ref. 218

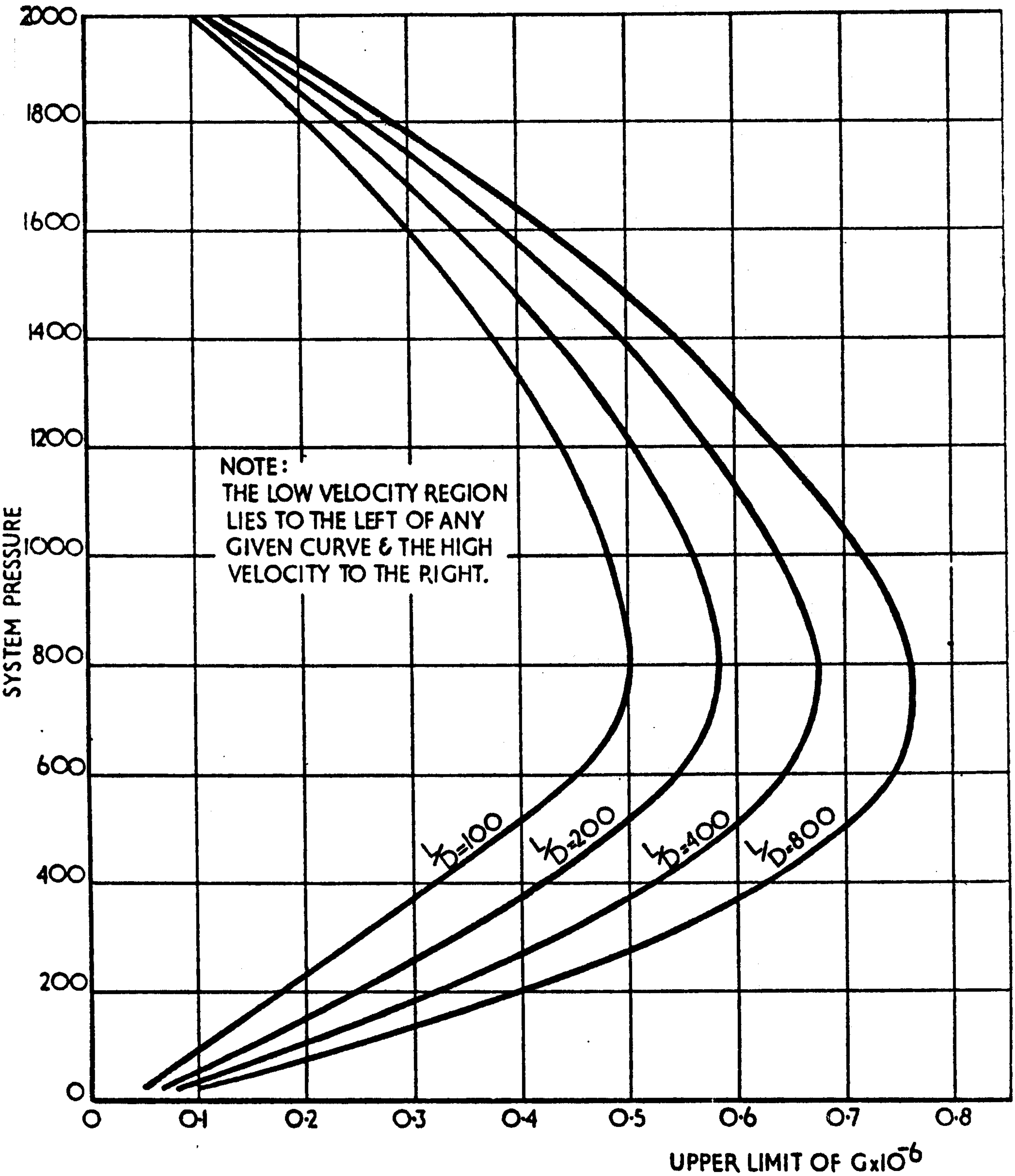
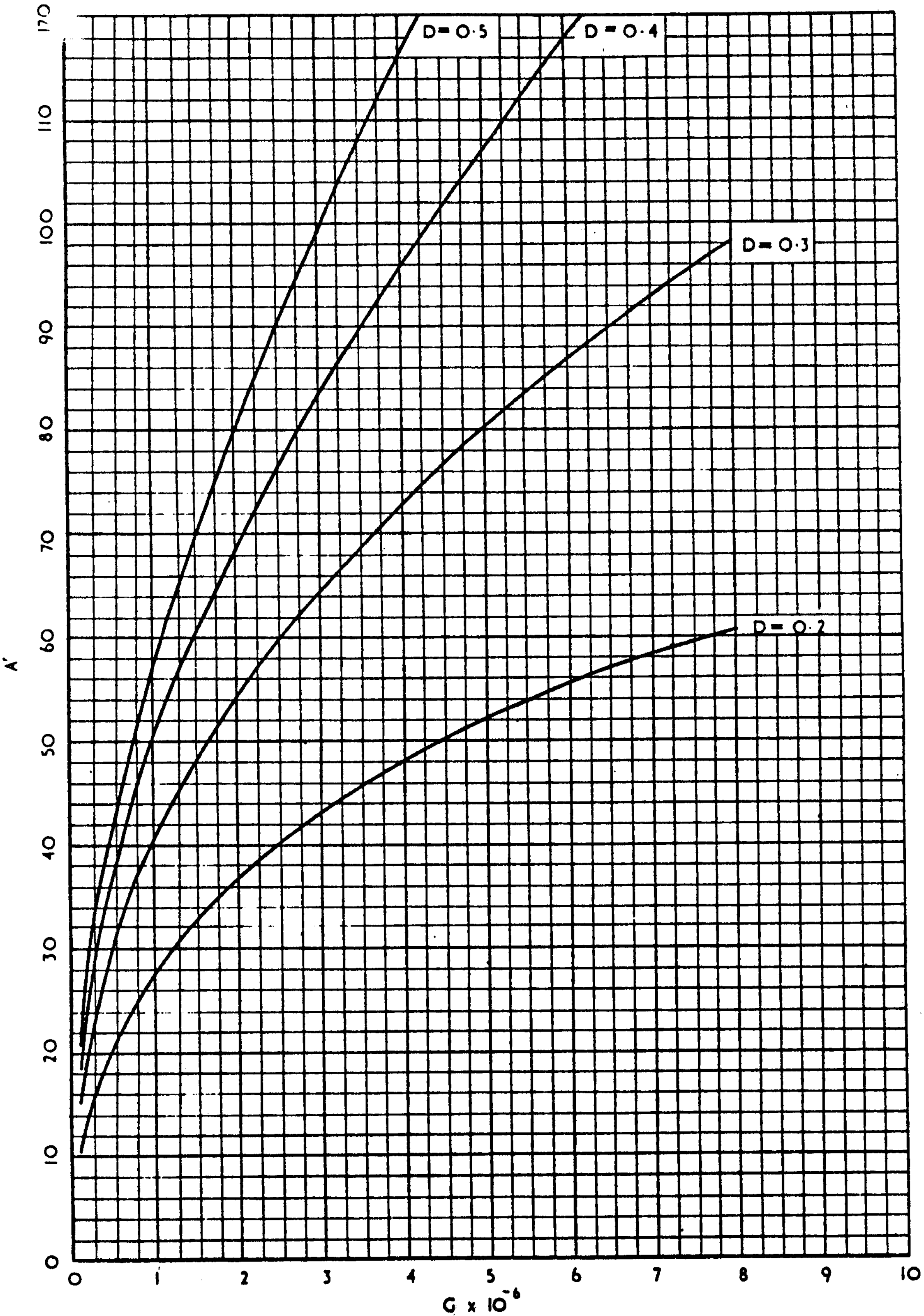


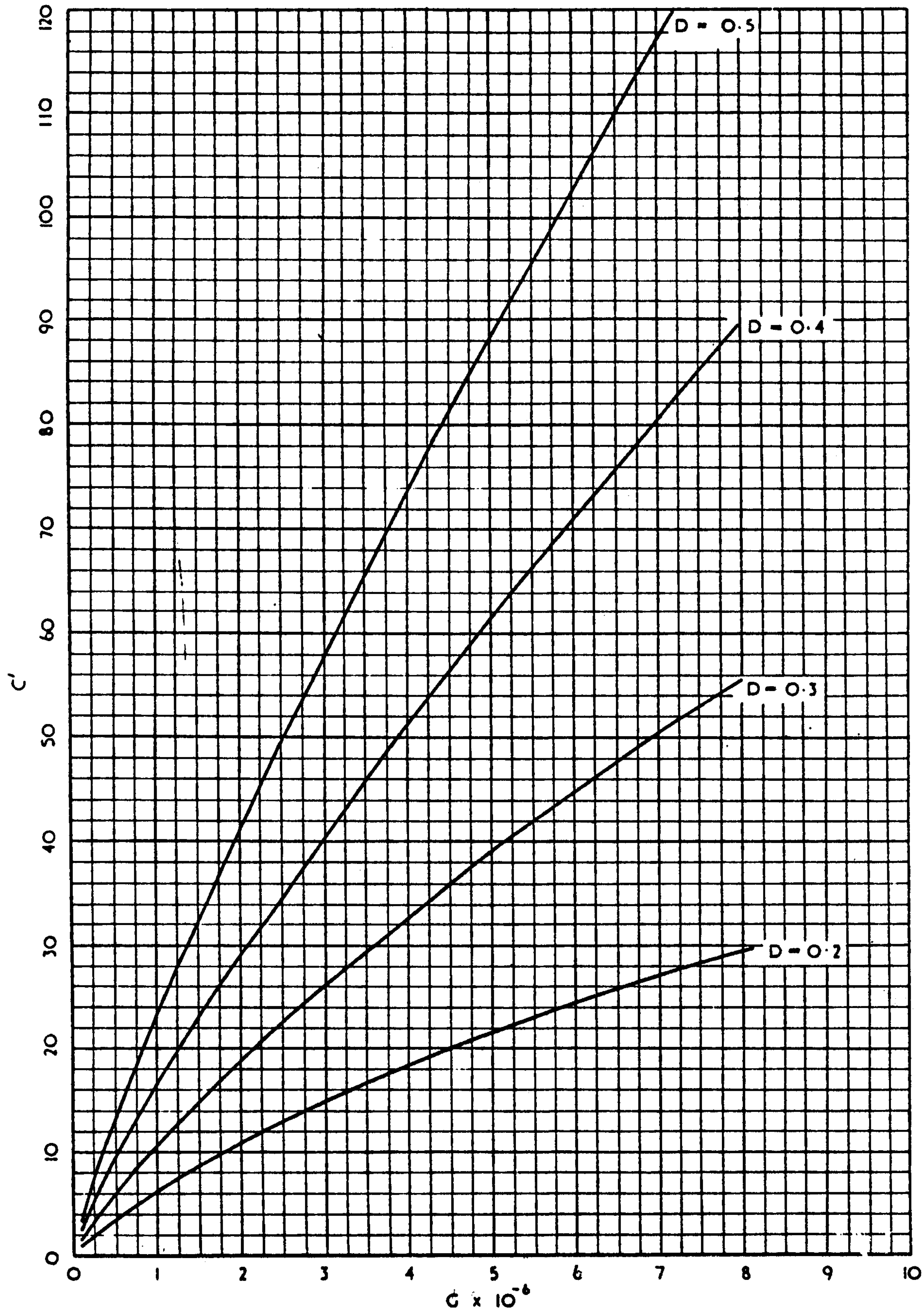
Figure 41. Approximate Limits of Velocity Regimes for Round Tubes

from Ref. 218



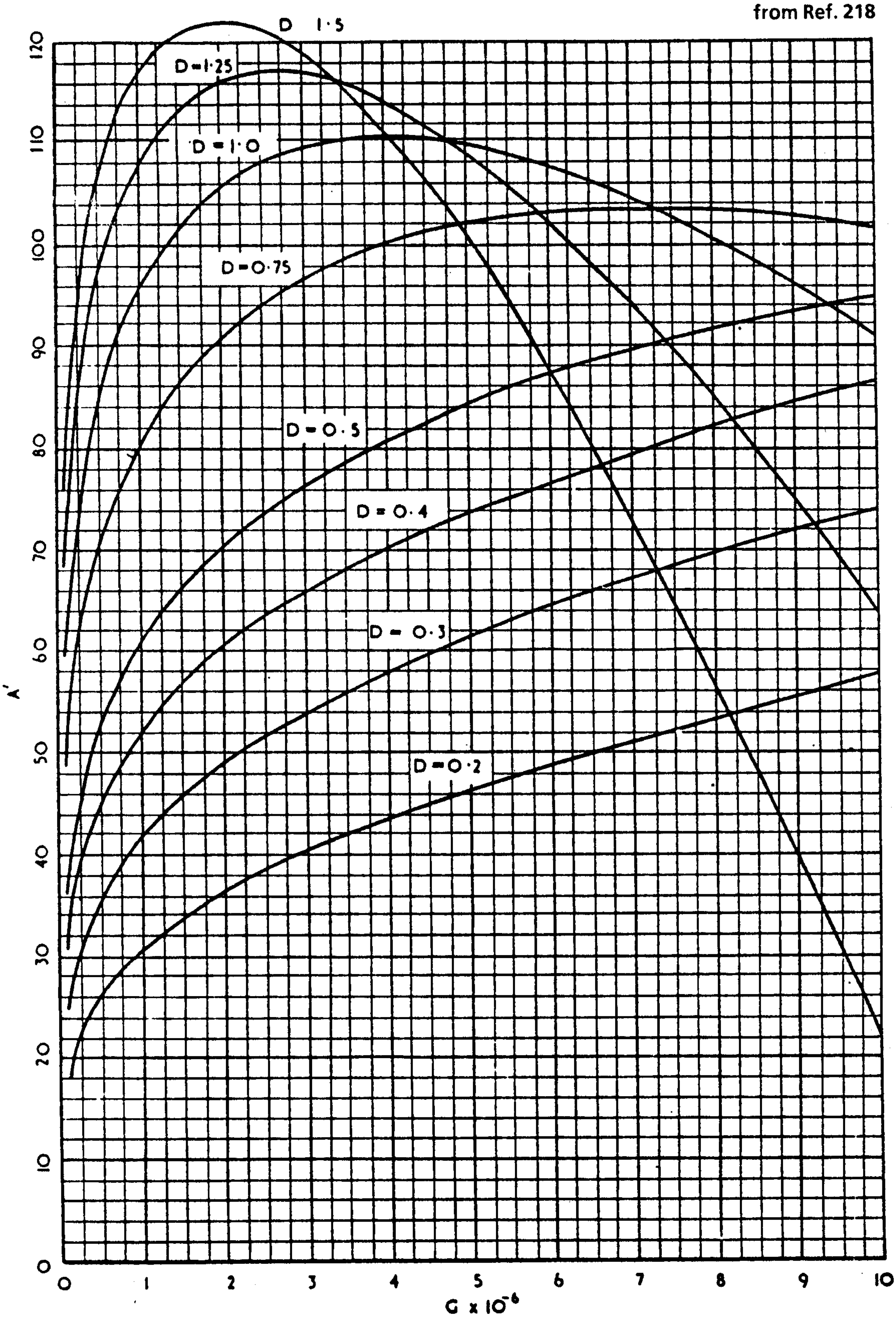
a. A′ for p = 560 psia

Figure 42. Constant Inputs for Thompson and Macbeth Correlation



b. C' for p = 560 psia

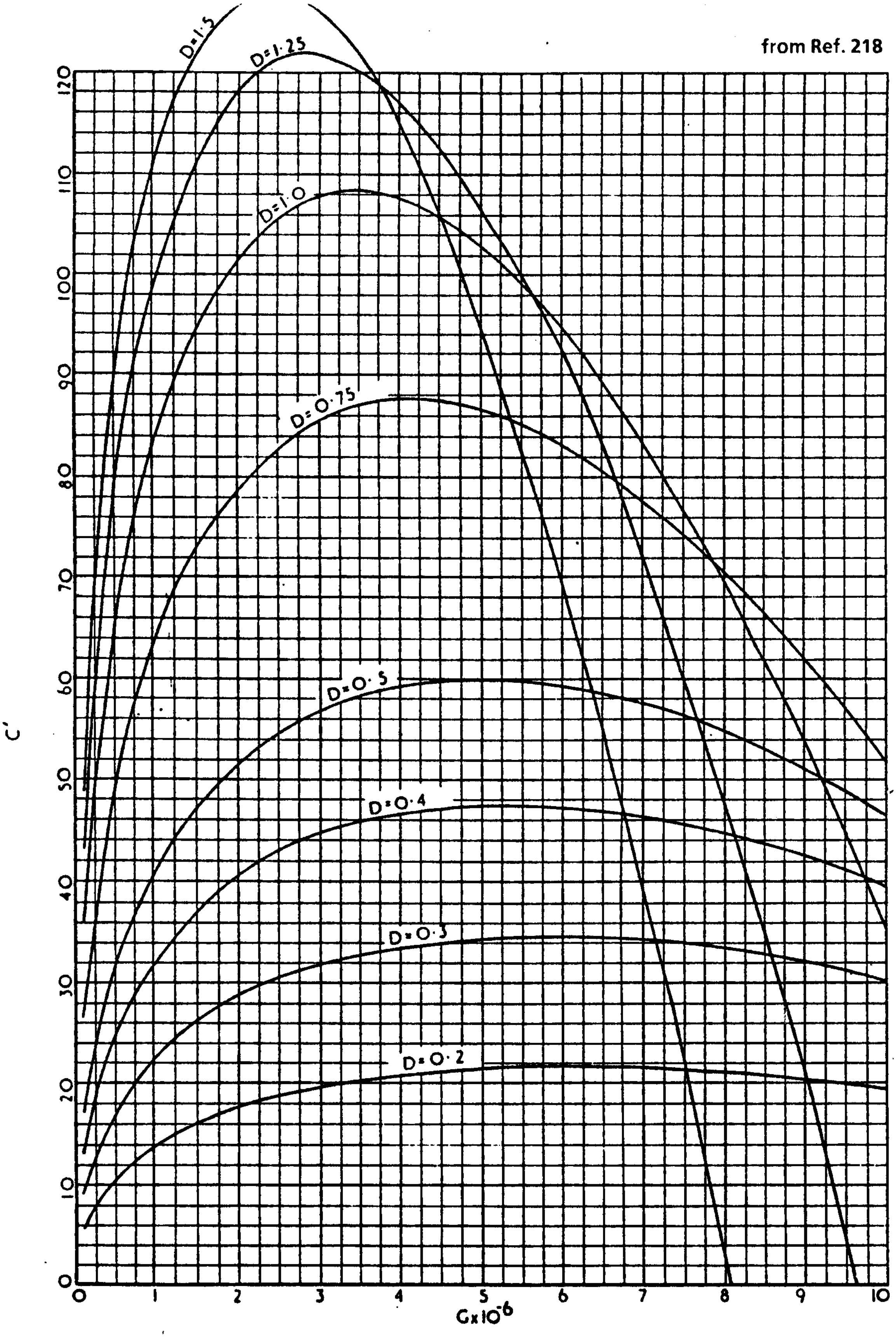
Figure 42. Continued



c. A' for p = 1000 psia

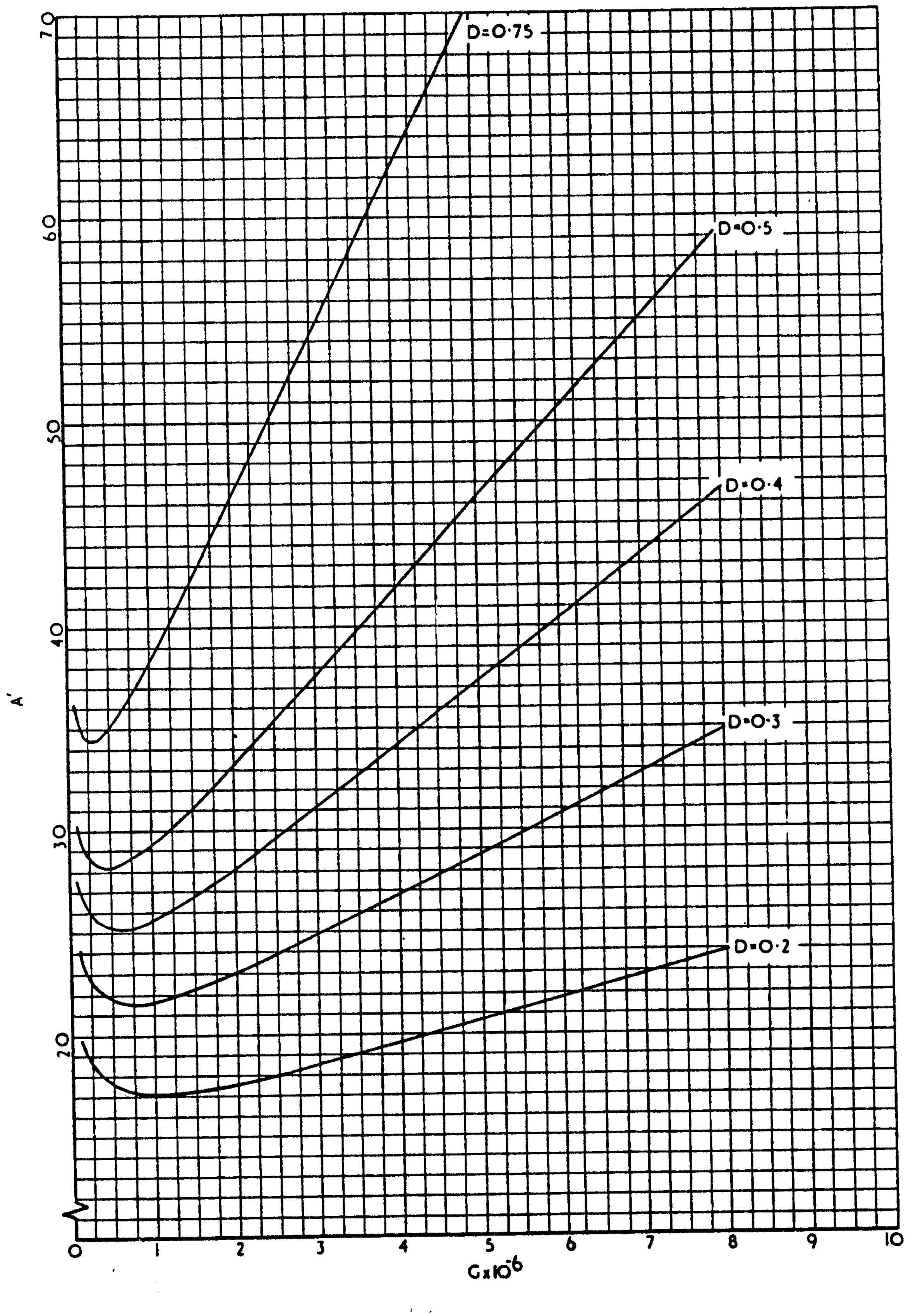
Figure 42. Continued

from Ref. 218



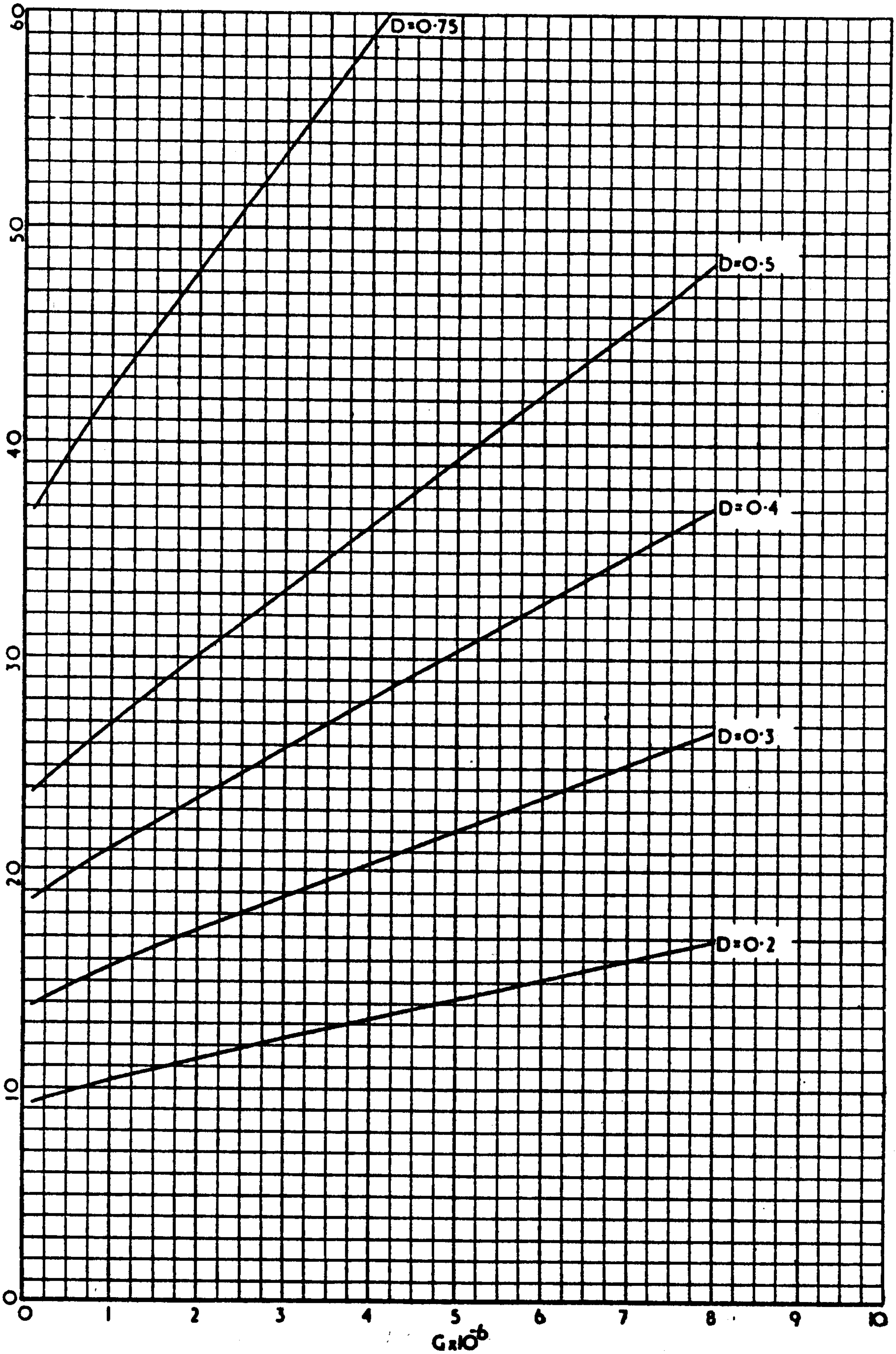
d. C' for p = 1000 psia

Figure 42. Continued



e. A' for p = 1550 psia

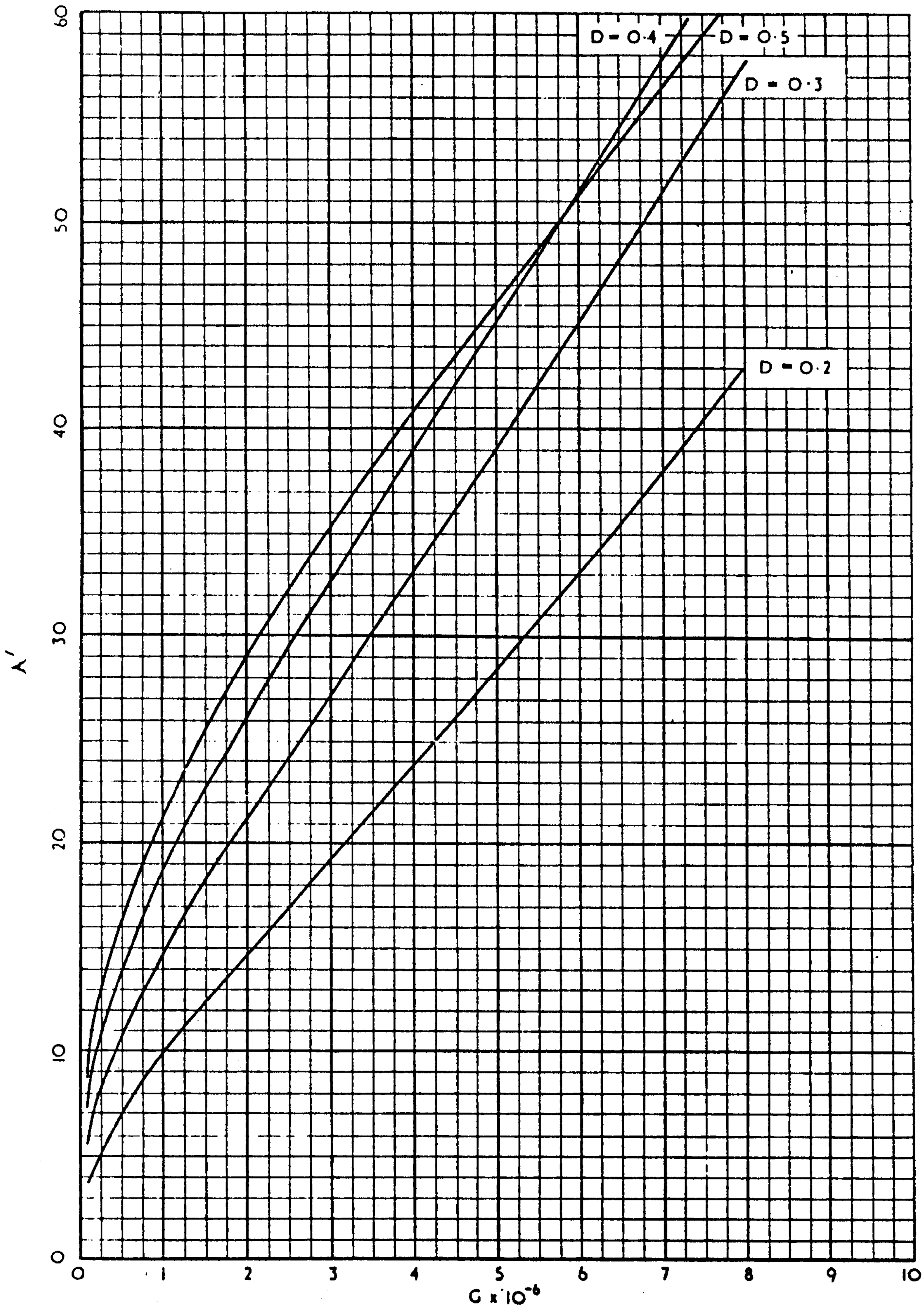
Figure 42. Continued



f. C' for p = 1550 psia

Figure 42. Continued

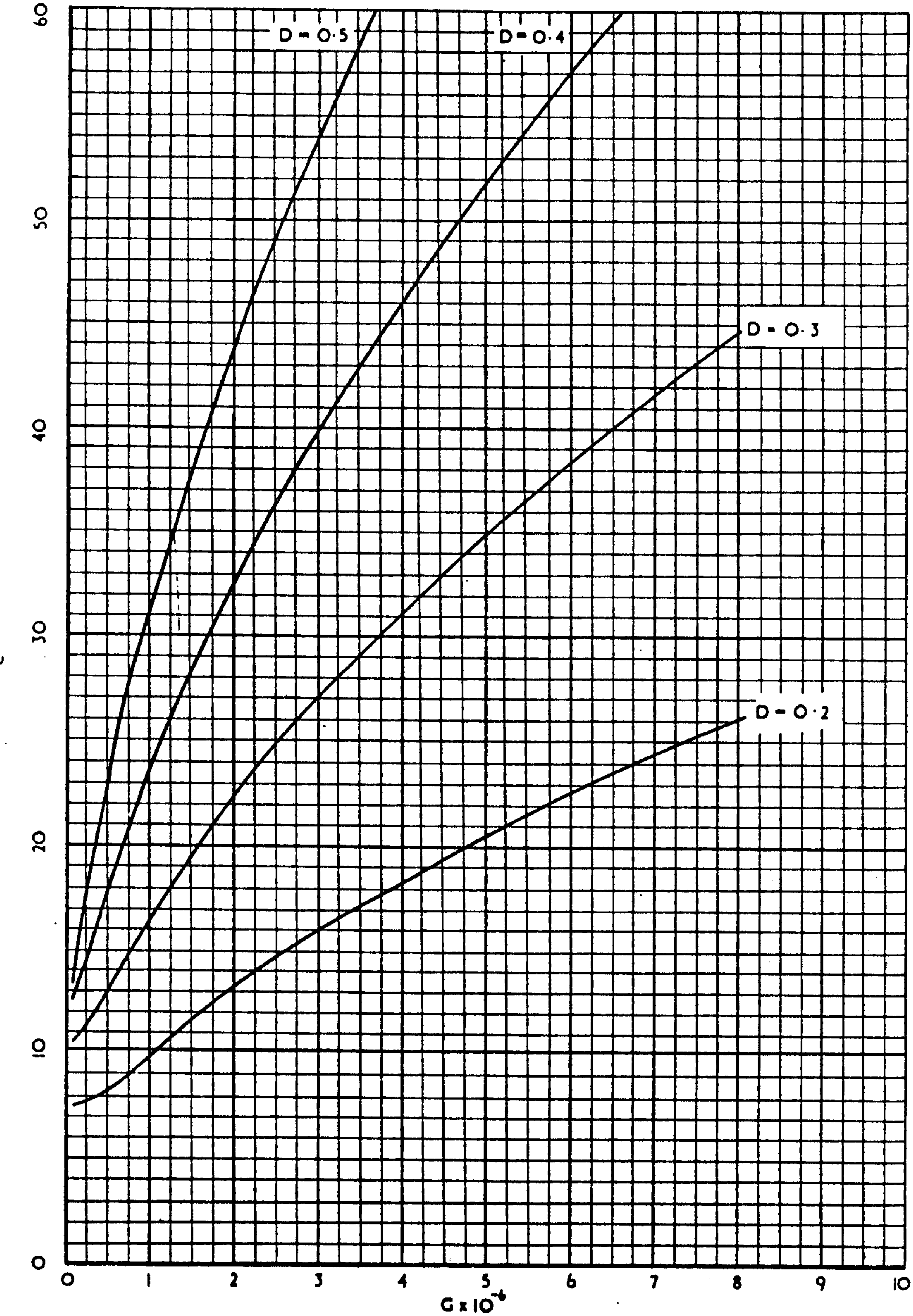
from Ref. 218



g. A′ for p = 2000 psia

Figure 42. Continued

from Ref. 218



h. C' for p = 2000 psia

Figure 42. Concluded

from Ref. 226

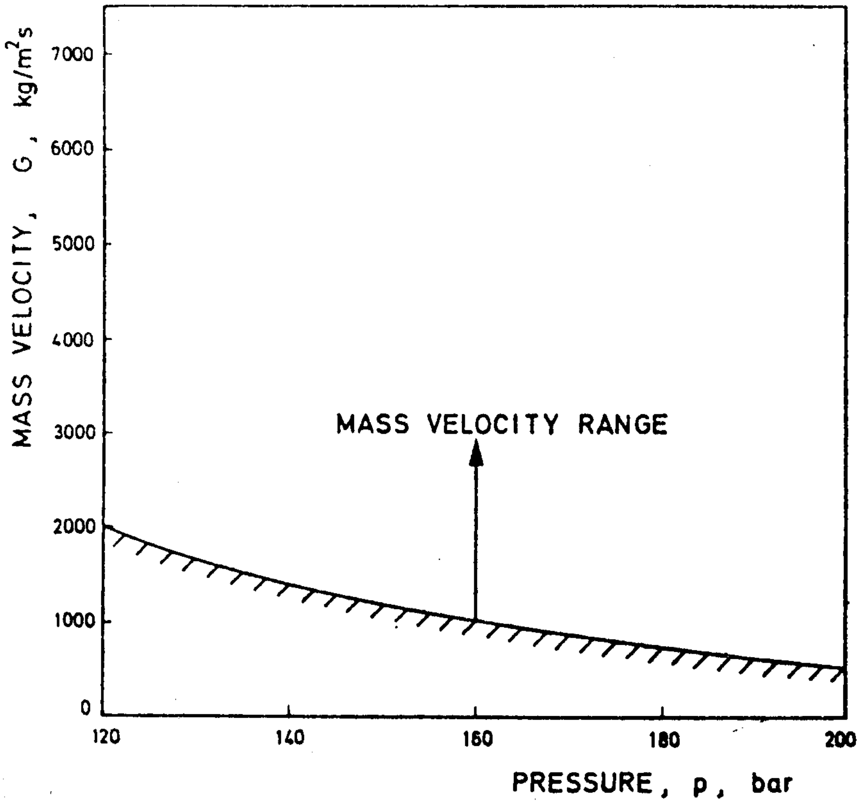


Figure 43. Mass Velocity Range for the Becker Burnout Correlation

Investigator	Ref	Yr	Organization	TYPE Subcool (surf)	Sat (bulk)	SURFACE Tube	Annulus	Rod / Wire	Plat/Chann	PARAMETER RANGE Tsub deg-F	L/D	Press psi	qdot [6] B/ft2s	Vel fps	mdot [7] lb/s	Q [8] gpm	G [9] lb/ft2s	Comments
Monrad & Pelton	253	42	Carnegie Inst	x			x			?	39,90	?	2-87		.4-2.5			counterflow
Davidson, et al	254	43	Con. Edison	x	x	x				203	211-722	500-2500	2028-10556	2-69	.09-1.43		20-311	tube bundles
Addoms	15	48	MIT	x	x			x		147	46	atm-2465	0-700	0-10				
Knowles	145	48	NRC, Canada	x			x			-72	7.6-76	40.6	0-684				158-618	
Kreith & Summerfield	90	49	JPL/Cal Tech	x		x				-300	30-73	16-202	105-403	6-13	.332-1.47			
McAdams, et al	24	49	MIT	x			x			-150	15	30-90	0-558	1-12				
Kaufman & Isely	255	50	NASA	x		x				-123	50	4.9-98	19.8-179		.054-.22			
Buchberg, et al	97	51	UCLA	x		x				236	111	100-2500	0-1050				267-2139	density study
Gunther	27	51	JPL/Cal Tech	x						-256	32,48	atm-164	117-3168	4.8-40				
McGill & Sibbitt	211	51	ANL/Purdue U	x		x				-615	21-168	265-3015	10-1244	.6-40				
Rohsenow & Clark	256	51	MIT	x		x				-250	52	1500,2000	0-972				up to 1556	
Jens & Lottes	257	52	Argonne N L	x	x	x			x	150	8.5-257	atm	0-83.3		.004-0.11			
Schweppe & Foust	80	53	UMI/Lehigh	x	x	x				?	8-19.5	?	10.5-57				100-1325	turbulence studied
Ellion	136	54	JPL/Cal Tech	x			x			100	1.3-8.3	16, 60	6-281	1.1, 5				
Piret & Isbin	258	54	U of MN		x	x					54	atm	.52-14.6	.77-3.51				
Clark & Rohsenow	23	54	MIT	x	x	x				00	52	100-2000	0-833				2.6-73	
Sabersky & Mulligan	259	55	Cal Tech	x		x				80	6.7	65-265	144-288		.73			friction evaluated
McEwen, et al	260	57	GE/AEC		x		x			to x=.56	656	1500	28-110				164-389	
Powell	69	57	JPL/Cal Tech	x	x	x				?	31-371	255-1075	0-360?				431-5395	
Reynolds	261	57	MIT	x	x	x				23	87	500-1500	0-794	5,10			239-592	
Zenkevich, et al	262	58	USSR	x	x	x				-72	40	2646-3087	?	2.1-15				tube bundles
Stein & Begell	115	58	Columbia U	x			x			?	160-480	?	11-313	4-45	.7-16.5			
Lowdermilk, et al	98	58	NASA	x	x	x				140	25-250	atm-100	0-3667	.1-98				
Griffith, et al	263	58	MIT, U of MI	x						-143	8	500-1500	6.9-750	20-38				void fraction measured
Core & Sato	264	58	Aerojet	x		x	x			-383	42	atm-500	0-744	1-22.7				outer wall heated
Gambill & Greene	265	58	Oak Ridge NL	x	x	x				58	2-20	94-930	0-15200	12-96				straight & swirl flow
Costello	238	59	U of WA	x			x			-103	35	16.7	0-228	1.5-4				vapor thickness measured
Foltz & Murray	266	59	Goodyear		x	x				0	21-495	75-105	.8-6.9				52.8-141	
Green, et al	31	59	Westinghouse	x	x	x			x	355	33-530	600-2000	33-1300				8.3-2156	
Zenkevich & Subbotin	267	59	USSR	x		x				-180	28-165	2058-3087	0-1192				171-1024	
Schrock & Grossman	268	59	U of CA		x	x				0	120,160	50-400	28-222				208-708	
Zenkevich, et al	269	60	USSR	x		x				-72	40	2646-3087	?	2.1-15				tube bundles
Owens & Schrock	270	60	U of CA	x		x				254	88-127	52-405	59-353				170-1090	tube bundle
Gambill, et al	113	61	Oak Ridge NL	x		x				260	45-70	29-220	222-10278	14.7-156				swirl flow; twisted tapes
Aladyev, et al	44	61	USSR	x	x	x				to x=-.4	19.5	300-3000	0-770				82-8200	
Mendler, et al	271	61	Westinghouse	x	x				x	109	68-149	800-2000	14-177				98-114	
Bankoff & Mason	272	62	Northwestern	x					x	-132	-	atm	?	.9-7.2				bubble growth; jet impingemnt
Swenson, et al	273	62	Babcock & Wil		x	x				0	176	3000	25-50				194,278	straight & swirl flow
Swenson, et al	118	62	Babcock & Wil		x	x				0	164	2000	19-156				139-361	
Styrikovich, et al	116	62	USSR	x	x	x				.5 to .9	27	1470-2646	0-692				82-410	
Schaefer & Jack	274	62	NASA	x		x				-343	28-61.5	69-307	0-11700	73-204			4500-12700	
Papell	275	63	NASA	x		x				0-263	21	37-179	53-230	3.8-12.5				
Judd & Wade	276	63	McMaster U	?	?		x			?	16-96	?	?	?				eccentricity effects
Becker	277	63	Sweden		x			x			64	36-156	44-106		.07-.73			rod clusters
Styrikovich, et al	278	63	USSR	x	x	x				.2 to 1	20-118	1470	0-872				82-410	
Tippets	28	64	GE	x	x				x	5-2598/0	17.6	995-1055	0-318				50-398	
Jiji & Clark	241	64	NYU, U of MI	x					x	-300	20	200-1000	39-417	1-6				flow temp probe measurement
Becker & Hernborg	86	64	Sweden	x			x			0-401	81	121-533	0-267				14.5-197	
Alekseev, et al	89	64	USSR	x	x	x	x	x		?	40,67	425-3000	0-970				0-1140	multiple flow configurations
Bergles & Rohsenow	157	64	MIT	x	x	x	x			-195	20-58	260	0-1667	0-57		0,3.6		
Jeglic & Yang	244	65	NASA	x	x	x				-195	100	3-100	0-234	2.2-8.25				flow oscillations measured
Hosler	279	65	Westinghouse	x	x				x	50	102	150-600	0-278				28-278	primarily photo data
Durant, et al	46	65	du Pont/SRL	x			x			-158	48-96	30-60	615-1000	10-28				knurled surfaces
Feldmanis	280	65	AFFDL		x	x				0	118	.5-3	?	.09-.72	.002-.003			low gravity flow
Stenning & Veziroglu	243	65	U of Miami	x		x				57	254	75	?		.007-.04			flow oscillations measured
Waters, et al	281	65	GE	x	x	x				-500	327	1015-1515	178-475				1322-1956	upstream burnout
Rousar & Van Huff	282	66	Aerojet/JPL	x		x				2-596	23,31	1000-3000	0-4032	38-150				
Ovhadi	283	66	OK State U	x	x	x				-108	245	atm	5.3-22.5		.02-.085			coiled flow
Bergles & Suo	284	66	MIT	x	x	x				-250	60-240	500-1000	0-crit				111-1111	
Brown	39	67	MIT				x			-180	20	30-90	0-140	0-5.5				bubble detector
Frost & Kippenhan	53	67	UT/UMA	x			x			-149	23	30	0-556	.8-4				varied surface tension
Weisman, et al	285	68	Westinghs/PNL	x				x		-230	28-59	1814-2314	0-178				139-833	rod bundles
Fiori & Bergles	20	68	MIT	x		x	x			-110	15-54	25-90	278-1528				139-2083	
Markels & Durfee	140	68	ARC, VA	x			x			-464	31-88	20-1500	0-475				42-556	applied a-c field
Lopina & Bergles	286	69	MIT,USAF Ac.	x		x				?	69-76	30-100	?		.028-.4			nonboiling, swirl flow
Borishanskiy & Fokin	62	69	USSR	x	x	x	x	x		-104	8.4-33	atm-356	0-431	0-8.2				unsteady heat application
Bergles & Dormer	171	69	MIT	x		x				-254	24-195	30-80	0-1528	5-60				
Zenkevich, et al	87	70	USSR	x	x	x	x			-108	93-730	570-2845	0-397				63-569	
Tolubinskiy & Kostanchuk	37	70	USSR	x					x	-108	?	atm-145	4.4-88	.26-.46				bubble growth measurements
Tolubinskiy, et al	137	70	USSR	x	x		x			x=-1to.3	67-1500?	725-2900	0-574				51-614	varied heating modes
Starman & Nekrasov	287	70	USSR		x	x				0	?	426-1421	44-927				171-696	
Stefanovic, et al	242	70	Yugoslavia		x	x	x			0?	27,69	atm	6-24		.57-.74			b.l. probe measurements
Osmachkin & Borisov	288	70	USSR	x	x	x				to x=.3	519	1015	63		15.2			
Saddis & Hall	289	70	U of Manc,Eng	x					x	-108	63	?	0-13	.33-1.6				drilled cavity
Johnson	290	70	U of CA,Berkl	x						-112	7.4	atm-2000	0-4	1,14				transient measurements
Kawamura, et al	291	70	Japan	x	x		x			-126	2.2	atm	0-1324	0-6.6				transient power application
Motley, et al	65	72	Westingh/CMU	x				x	x	?	40,227	163-2400	0-400				417-1111	rod bundle
Mattson	292	72	U of MI	x					x	-119	?	105-352	7.5-85				322-1139	
Becker, et al	226	72	Royal I, Swe	x		x				-522	200-500	1450-2900	12-312				32-1548	
Ahmad & Groeneveld	64	72	AECL, Can	x			x			7-18 B/#	155,222	90-240	0-317				28-667	
Abdelmessih, et al	26	72	U of Toronto	x						3.5	2.7?	?	16.5-40.5	3-7.5				
Lopina & Bergles	111	73	MIT	x		x				100	d=.194 in	30-50	0-972	9.6-25				swirl flow
Plummer, et al	293	73	GE		x	x				.3 to 1	445	100-1000	0-243				13.9-69	fouling evaluated, film boiling
Purcupile, et al	66	73	Westingh/CMU	x		x				2-51 B/#	90	157-362	0-107				287-1803	
Tolubinskiy & Matorin	61	73	USSR	x		x				?	10-15	48-191	0-1192	8.2-33				binary mixes
Megerlin, et al	114	74	WGr,Rayth,ISU	x		x				47-365	9.5	29-252	0-2778				428-817	tube inserts
Katto & Monde	294	74	Japan		x					0	-	atm	0-1589	0-197				jet impingement
Hughes	21	74	Penn State	x						0-110	11	110-150	0-172	1-4.4				straight and curved flow
Levitan & Lantsman	223	75	USSR	x	x	x				?	188-375	711-2697	0-243				205-614	
Iloeje, et al	295	75	MIT		x	x				.3 to 1	8.1	1000	0-243				13.9-69	transient data
Kovalev, et al	47	76	USSR	x				x		-126	-	atm	0-914	.07-.46				finned geoms; electron bomb htg
Spencer & Rousar	296	77	NASA	x		x				?	3.6-204	2500-5000	173-7920				4915-25000	
Cumo, et al	297	77	Italy	x						-131	20	155-316	5.3-25				28.7-75.8	vapor thickness measured

Investigator	Ref	Yr	Organization	TYPE: Subcool (surf)	TYPE: Sat (bulk)	SURFACE: Tube	SURFACE: Annulus	SURFACE: Rod / Wire	SURFACE: Plate, Channel	sub -F	PARAMETER RANGE: L/D	Press psi	qdot [6] B/ft2s	Vel fps	mdot [7] lb/s	Q [8] gpm	G [9] lb/ft2s	Comments
Eddington & Kenning	298	78	Oxford U	x					x		?	17	0-18.5	.49-1				Del Valle/Kenning rig
Eddington, et al	299	78	Oxford U/USSR	x					x		?	17	0-9.7	.49				Del Valle/Kenning rig
Cheng, et al	122	78	U of Ottawa	x	x	x					8.5	atm	0-177				14-42	
Kutateladze & Malenkov	300	78	USSR		x		x				?	?	0	?				bubble stability
Monde & Katto	301	78	Japan	x	x				x		-	atm	0-1750	6.7-85				impinging jet
Yucel & Kakac	138	78	Turkey	x					x	6	14-28	atm	0-530				256-1280	varied heater orientation
Drizius, et al	302	78	USSR	x		x				19	40-153	57-194	0-7176				819-5120	swirl flow
Doroschuk, et al	303	78	USSR	x	x	x	x				225,?	290-2422	0-106				154-614	
Ragheb, et al	304	78	U of Ottawa	x	x				x	4	8.3	atm	0-88				7-21	transition studied
Mel'nikov, et al	105	79	USSR	x	x		x			io.15	?	435	0-468				512-717	fouling studied
Chen, et al	305	79	U of Ottawa	x		x				4	202,253	atm	0-230				20-82	transient results
Ragheb & Cheng	306	79	U of Ottawa	x	x				x	4	8.3	atm	0-887				7-21	
Bennett & Chen	307	80	Lehigh U		x	x					3	?	62-265				33-328	pure and binary coolant mixture
Mori, et al	308	80	Japan	x				x	x		75	atm	0-132	.016-.023				bubble generation
Petukhov, et al	309	80	USSR		x	x				?	300	atm	.0009-.37				3.1-68	
Koba, et al	310	80	USSR	x	x	x					15-200	atm-160	353-1854	1.6-98				small diameter tubes
Tolubinskiy, et al	249	80	USSR	x			x				?	atm-72.5	0-265	1.6-6.6				acoustic measurements
Yilmaz & Westwater	311	80	HTRI/U of IL		x	x					20	atm-16.6	0-66	0-22				
Ornatskiy, et al	117	80	USSR	x			x			3	25-300	1066-2916	0-600				102-717	
Ornatskiy & Sharayevskiy	248	80	USSR	x			x			6	gap=.5mm	atm-58	?	3.3-8.2				acoustic measurements
Leontiev, et al	119	81	USSR	x	x	x				o.7	69	995-1991	0-640				154-410	
Zeigarnik, et al	93	81	USSR	x					x	8	54	73-435	706-5482				819-5120	
Yao	312	81	Carnegie-Mell	x	x		x				7-136	atm-16.5	0-8	.41-.66			39-62	gaps
Styrikovich, et al	313	81	USSR	x	x	x				to .9	75,152	995-1989	0-397				205,410	
Ragheb, et al	121	81	U of Ottawa	x	x	x				4	d=1.2 cm	atm	0-221				13.9-41.6	ss & transient heating
Ramanchan, et al	314	81	India	x			x			5	150	?	6.9-209	.5-1.8				
Jensen & Bergles	92	81	U of WI,IA St	x	x	x					83-174	136	4.8-70.6				117-1120	straight & coiled flow
France, et al	315	81	Argonne N L	x		x					1297	1015-2219	44-141				143-655	
Agrawal, et al	316	82	India		x	x					210	?	.7-1.8				40-79	plain & swirl flow
Aounallah, et al	317	82	Oxford U		x	x					48	atm-20	?				21.5,41.6	
Del Valle & Kenning	79	82	Oxford U	x					x	4	21	atm	26.5-413	2.6-6.6				Eddington et al rig
Fukuyama & Hirata	318	82	Japan	x		x					51	51-216	0-794				8540-10834	questionable mass flux
Katto & Ashida	319	82	Japan	x	x	x				kJ/kg	50	284-497	0-73				143-1434	
Leung, et al	43	82	AECL,Can/UCA	x			x				1162	1407	0-110				410-1024	different heater configurations
Nishima & Ishii	320	82	Japan/ANL	x	x		x			?	108	atm	?				1-7.2	
Nishikawa, et al	188	82	Japan	x	x	x				kJ/kg	154	493-696	0-8.1				41-266	
Papaioannou & Koumoutsos	59	82	Greece		?			x			-	?	0-2.2	0-.33				bubble generation
Rogers, et al	239	82	U of Ottawa	x	x		x			0	28-54	22.6	13-247				12.3-246	void fraction measurements
Katto & Yokoya	32	82	Japan	x	x	x				kJ/kg	200,333	284-497	0-44				225-1843	
Sekoguchi, et al	321	82	Japan	x	x	x				to.04	91-154	29-232	10.6-164				61-410	
Viannay & Karian	322	82	France	x			x		x		?	?	8.8-883	2.3-36				
Wei & Maa	60	82	Taiwan				x				10.5-32	?	0-79	.1-1				surface tension effect
Ueda, et al	323	83	Japan	x	x	x				to.62	6,10	40.6	0-48.5				73-300	
Sharon, et al	324	83	Northwestern	x	x		x				60,100	19-102	0-44				68,117	
Bilicki	54	83	Poland		x	x					38	87	0-9.7				127-246	
Kliamenko & Sudarchikov	325	83	USSR	x	x	x					185	26-102	.9-3.8				35-154	large length
Kalayda, et al	101	83	USSR	x			x			72	282	1160-2320	88-225				102-410	
Baines, et al	326	84	MIT	x	x				x		-	atm	0-143	3.3-16.4				gravity & jet driven films
Katto & Ohno	187	84	Japan	x		x				kJ/kg	100	284-499	0-18				25-430	
Kozhelupenko, et al	327	85	USSR	x			x				33-50	atm-145	4.4-141	.07-2.3				
Hino & Ueda	328	85	Japan	x			x				40	21.3	0-17.7				105-254	fluid temp probe survey
Evans, et al	329	85	Lehigh U		x	x					88	35-83	1.6-5.1				2.7-17.4	vapor probe measurements
Edelman, et al	240	85	Technion, Isr	x		x				.49	167,169	atm	2.6-21				16.4-51.2	void fraction measurements
Boyd, et al	94	86	Sandia NL	x		x					116	232	132-375				129-717	
Fujita, et al	330	86	Japan		x	x					4.8	atm-72.5	0-8.8				?	tube bundles
Fukuyama, et al	331	86	Japan	x		x				94	50	84-345	0-895				5734-11469	questionable mass flux
Grigoriev, et al	332	86	USSR		x	x			x		?	atm-290	.53-13.2					
Muller-Steinhagen, et al	333	86	Canada	x			x	x		50	7,625	atm-114	.07-35				1.1-146	heater geometry varied
Tolubinskiy & Domashev	234	86	USSR		x?	x?				?	100	1421-2132	0-485				205	CHF detector
Ueda & Kim	334	86	Japan/Korea	x			x				40	21.3	0-39.7				53-254	
Aounallah & Kenning	335	87	Oxford U	x		x					52	16	0-15				21.5-42	
Yezhova, et al	336	87	USSR	x		x				to.2	D=8mm	1422-2421	0-468				205-758	
Lee & Shen	337	87	U of Ottawa	x		x	x				dh=25 mm	?	0-132				10-41	falling films also
Inasaka & Nariai	338	87	Japan	x		x				=-.14	3-100	atm	?				1434-4096	
El-Genk, et al	339	88	U of NM	x			x			5	39-68	17.1	13.7-125				0-53.2	
Muller-Steinhagen, et al	340	88	N Zeal,Can	x			x			2?	7	atm-106	.07-33				41-156	
Passos & Gentile	441	88	France	x		x					6.25	21.8-58	0-53				82-266	transition evaluation
Lee & Simon	342	89	Motorola,UMN	x					x	83	.05-.65	18.7-40	0-150	8.1-28				
Celata, et al	122	89	Italy	x	x	x				4	153-299	181-399	0-10.6				202-312	transient & s.s. heating
Boyd	172	89	Sandia NL	x		x					97	241	0-3707				901-6554	
Jung, et al	343	89	UMD,NIST,NBS	x	x	x				x=.95	444	38-58	.88-4		.035-.1		51-147	
Gu, et al	108	89	U of KY, APPL	x					x	8.4	1	atm	0-124	3.3-13				straight & curved flow
Polonskii, et al	344	90	USSR		?	x					?	1450-2030	17.7-115				102-410	
Hasan, et al	345	90	AZ State,EPRI	x			x				121	35-45	.9-13.2				119-226	
Cornwell	346	90	UK		x	x					?	atm	0-3.4				19.5	tube bundles
Sivagnanam & Varma	347	90	India	x				x			1650	?	18-265	.5-3.3				binary mixtures
Dietrich, et al	348	90	New Zealand	x	x			x			-	atm	.04-44	.03-1.3				coiled wires
Feng & Johannsen	349	91	Germany	x		x					5.2	16-174	1.8-291				2-205	transition boiling
Celata, et al	350	91	Italy	x	x	x				4	153-299	181-399	0-10.6				202-312	trans & s.s. heating,press,flow

NOTES:

[1] coolant treatment
- dis — distilled
- dei — deionized
- dem — demineralized
- deg — degassed
- gas — gassed
- untr — untreated
- nwtd — no water treatment discussed

[2] dTsub — subcooling, Tsat-Tbulk

[3] Vel — velc

[4] L/D — for

for

for

for

[5] Dh — hydr